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Specification and Drawings, as originally filed, with Application for Patent Serial No: **2,436,327,** on July 31, 2003, by **NATIONAL STEEL CAR LIMITED,** assignee of James W. Forbes, for "Rail Road Car Truck and Bearing to Sideframe Interface Therefor".

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RAIL ROAD CAR TRUCK AND BEARING TO SIDEFRAME INTERFACE THEREFOR

Abstract of the Disclosure

A rail road freight car truck is provided that has a truck bolster and a pair of side frames, the truck bolster being mounted transversely relative to the side frames. The mounting interface between the ends of the axles and the sideframe pedestals allows lateral rocking motion of the sideframes in the manner of a swing motion truck. The lateral swinging motion is combined with a longitudinal self steering capability, either from the use of a lateral rocker in combination with an elastomeric bearing adapter pad, or by use of a longitudinally oriented rocker that may tend to permit resistance to self steering that is proportional to the weight carried across the interface.

RAIL ROAD CAR TRUCK AND BEARING TO SIDEFRAME INTERFACE THEREFOR

Field of the Invention

This invention relates to the field of rail road cars, and, more particularly, to the field of three piece rail road car trucks for rail road cars.

Background of the Invention

Rail road cars in North America commonly employ double axle swivelling trucks known as "three piece trucks" to permit them to roll along a set of rails. The three piece terminology refers to a truck bolster and pair of first and second sideframes. In a three piece truck, the truck bolster extends cross-wise relative to the sideframes, with the ends of the truck bolster protruding through the sideframe windows. Forces are transmitted between the truck bolster and the sideframes by spring groups mounted in spring seats in the sideframes. The sideframes carry forces to the sideframe pedestals. The pedestals seat on bearing adapters, whence forces are carried in turn into the bearings, the axle, the wheels, and finally into the tracks.

One general purpose of a resilient suspension system may tend to be to reduce force transmission to the car body, and hence to the lading. This may apply to very stiff suspension systems, as suitable for use with coal and grain, as well as to relatively soft suspension systems such as may be desirable for more fragile goods, such as rolls of paper, automobiles, shipping containers, fruit and vegetables, and white goods.

Ride quality can be judged on a number of different criteria. There is longitudinal ride quality, where, often, the limiting condition is the maximum expected longitudinal load experienced during humping or flat switching, or slack run-in and run-out. There is vertical ride quality, for which vertical force transmission through the suspension is the key determinant. There is lateral ride quality, which relates to the lateral response of the suspension. There are also other phenomena to be considered, such as truck hunting, the ability of the truck to self steer, and, whatever the input perturbation may be, the ability of the truck to damp out undesirable motion. These phenomena tend to be inter-related, and the

optimization of a suspension to deal with one phenomenon may yield a system that may not necessarily provide optimal performance in dealing with other phenomena.

In terms of optimizing truck performance, it may generally be desirable to obtain a measure of self steering in the truck, desirable to avoid truck hunting, and desirable to have a relatively soft lateral and vertical response. It would be advantageous to be able to obtain the desirable relatively soft dynamic response to lateral and vertical perturbations, to obtain a measure of self steering, and yet to maintain resistance to lozenging (or parallelogramming). Lozenging, or parallelogramming, is non-square deformation of the truck bolster relative to the side frames of the truck, entailing relative rotation of the truck bolster to the side frames about a vertical axis of rotation, namely warp deformation of the truck. That is, when the truck deflects too easily to a parallelogram condition, it may also have an undesirably low hunting threshold. However, the very concept of self steering may tend to require some permissible level of angular mis-orientation of one axle relative to the other, which, in turn, implies a measure of angular deflection of at least one axle relative to the side frames.

With the foregoing in mind, one determinant of overall ride quality is the dynamic response to lateral perturbations. That is, when there is a lateral perturbation at track level, the rigid steel wheelsets of the truck may be pushed sideways relative to the car body. Lateral perturbations may arise for example from uneven track, or from passing over switches or from turnouts and other track geometry perturbations. When the train is moving at speed, the time duration of the input pulse due to the perturbation may be very short. The suspension system of the truck reacts to the lateral perturbation. It is generally desirable for the force transmission to be relatively low. High force transmissibility, and corresponding high lateral acceleration, may tend not to be advantageous for the lading. This is particularly so if the lading includes relatively fragile goods.

In general, leaving aside, for now, the lateral stiffness of the bearing to bearing adapter and pedestal roof interface, for the purposes of analysis, the lateral stiffness of the suspension may tend to reflect the combined displacement of (a) the sideframe between (i) the bearing adapter and (ii) the bottom spring seat (that is, the sideframes swing laterally as a pendulum with the pedestal bearing adapter being the top pivot point for the pendulum); (b) the lateral deflection of the springs between (i) the lower spring seat in the sideframe and (ii) the upper spring mounting against the underside of the truck bolster, and (c) the moment and the associated transverse shear force between (i) the spring seat in the sideframe and (ii) the upper spring mounting against the underside of the truck bolster.

In a conventional rail road car truck, the lateral stiffness of the spring groups is sometimes estimated as being approximately ½ of the vertical spring stiffness. Thus the choice of vertical spring stiffness may strongly affect the lateral stiffness of the suspension. The vertical stiffness of the spring groups may tend to yield a vertical deflection at the releasable coupler from the light car (i.e., empty) condition to the fully laden condition of about 2 inches. For a conventional grain or coal car subject to a 286,000 lbs., gross weight on rail limit, this may imply a dead sprung load of some 50,000 lbs., and a live sprung load of some 220,000 lbs., yielding a spring stiffness of 25 – 30,000 Lbs./in., per spring group (there being, typically, two groups per truck, and two trucks per car). This may yield a lateral spring stiffness of 13 – 16,000 Lbs./in per spring group. It should be noted that the numerical values given in this background discussion are approximations of ranges of values, and are provided for the purposes of general order-of-magnitude comparison, rather than as values of a specific truck.

The second component of stiffness relates to the lateral deflection of the sideframe itself. In a conventional truck, the weight of the sprung load can be idealized as a point load applied at the center of the bottom spring seat. That load is carried by the sideframe to the pedestal seat mounted on the bearing adapter. The vertical height difference between these two points may be in the range of perhaps 12 to 18 inches, depending on wheel size and sideframe geometry. For the general purposes of this description, for a truck having 36 inch wheels, 15 inches (+/-) might be taken as a roughly representative height.

The pedestal seat may typically have a flat surface that bears on an upwardly crowned surface on the bearing adapter. The crown may typically have a radius of curvature of about 60 inches, with the center of curvature lying below the surface (*i.e.*, the surface is concave downward). When a lateral shear force is imposed on the springs, there is a reaction force in the bottom spring seat that will tend to deflect the sideframe, somewhat like a pendulum. When the sideframe takes on an angular deflection in one direction, the line of contact of the flat surface of the pedestal seat with the crowned surface of the bearing adapter will tend to move along the arc of the crown in the opposite direction. That is, if the bottom spring seat moves outboard, the line of contact will tend to move inboard. This motion is resisted by a moment couple due to the sprung weight of the car on the bottom spring seat, acting on a moment arm between (a) the line of action of gravity at the spring seat and (b) the line of contact of the crown of the bearing adapter. For a 286,000 lbs. car the apparent stiffness of the sideframe may be of the order of 18,000 – 25,000 Lbs./in, measured at the bottom spring

seat. That is, the lateral stiffness of the sideframe (i.e., the pendulum action by itself with a bearing adapter with a 60" crown radius) can be greater than the already relatively high lateral stiffness of the spring group in shear, and this apparent stiffness is proportional to the total sprung weight of the car, including lading. When taken as being analogous to two springs in series, the overall equivalent lateral spring stiffness may be of the order of 8,000 to 10,000 Lbs./in., per sideframe. A car designed for lesser weights may have softer apparent stiffness. This level of stiffness may not always yield as smooth a ride as may be desired.

There is another component of spring stiffness due to the unequal compression of the inside and outside portions of the spring group as the bottom spring seat rotates relative to the upper spring group mount under the bolster. This stiffness, which is additive to (that is, in parallel with) the stiffness of the sideframe, can be significant, and may be of the order of 3000 - 3500 Lbs./in per spring group, depending on the stiffness of the springs and the layout of the group. Other second and third order effects are neglected for the purpose of this description. The total lateral stiffness for one sideframe, including the spring stiffness, the pendulum stiffness and the spring moment stiffness, for a S2HD 110 Ton truck may be about 9200 Lbs./inch per side frame.

It has been observed that it may be preferable to have springs of a given vertical stiffness to give certain vertical ride characteristics, and a different characteristic for lateral perturbations. In particular, a softer lateral response through the main spring groups may be desired at high speed (greater than about 50 m.p.h.) and relatively low amplitude to address a truck hunting concern, while a different spring characteristic may be desirable to address a low speed (roughly 10 – 25 m.p.h.) roll characteristic, particularly since the overall suspension system may have a roll mode resonance lying in the low speed regime.

An alternate type of three piece truck is the "Swing Motion" truck. One example of a swing motion truck is shown at page 716 in the *1980 Car and Locomotive Cyclopedia* (1980, Simmons-Boardman, Omaha). In a swing motion truck, the sideframe is mounted as a "swing hanger" and acts much like a pendulum. In contrast to the truck described above, the bearing adapter has an upwardly concave rocker bearing surface, having a radius of curvature of perhaps 10 inches and a center of curvature lying above the bearing adapter. A pedestal rocker seat nests in the upwardly concave surface, and has itself an upwardly concave surface that engages the rocker bearing surface. The pedestal rocker seat has a radius of curvature of perhaps 5 inches, again with the center of curvature lying upwardly of the rocker.

In this instance, the rocker seat is in dynamic rolling contact with the surface of the bearing adapter. The upper rocker assembly tends to act more like a hinge than the shallow crown of the bearing adapter described above. As such, the pendulum may tend to have a softer, perhaps much softer, response than the analogous conventional sideframe. Depending on the geometry of the rocker, this may yield a sideframe resistance to lateral deflection in the order of ¼ (or less) to about ½ of what might otherwise be typical. If combined in series with the spring group stiffness, it can be seen that the relative softness of the pendulum may tend to become the dominant factor. To some extent then, the lateral stiffness of the truck becomes less strongly dependent on the chosen vertical stiffness of the spring groups at least for small displacements. Furthermore, by providing a rocking lower spring seat and a transom the swing motion truck may tend to reduce, or to eliminate, the component of lateral stiffness that may tend to arise because of unequal compression of the inboard and outboard members of the spring groups, thus further softening the lateral response.

For the purposes of rapid estimation of truck lateral stiffness, the following formula can be used:

$$k_{truck} = 2 \times [\ (k_{sideframe})^{-1} + (k_{spring\ shear})^{-1}]^{-1}$$

where

$k_{sideframe}$ = $[k_{pendulum} + k_{spring\ moment}]$

$k_{spring\ shear}$ = The lateral spring constant for the spring group in shear.

$k_{pendulum}$ = The force required to deflect the pendulum per unit of deflection, as measured at the center of the bottom spring seat.

$k_{spring\ moment}$ = The force required to deflect the bottom spring seat per unit of sideways deflection against the twisting moment caused by the unequal compression of the inboard and outboard springs.

In a pure pendulum, the relationship between weight and deflection is approximately linear for small angles of deflection, such that, by analogy to a spring in which $F = kx$, a lateral constant (for small angles) can be defined as $k_{pendulum} = W / L$, where k is the lateral constant, W is the weight, and L is the pendulum length. Further, for the purpose of rapid comparison of the lateral swinging of the sideframes, an approximation for an equivalent pendulum length for small angles of deflection can be defined as $L_{eq} = W / k_{pendulum}$. In this equation W represents the sprung weight borne by that sideframe, typically ¼ of the total sprung weight for a symmetrical car. For a conventional truck, L_{eq} may be of the order of about 3 or 4 inches. For a swing motion truck, L_{eq} may be of the order of about 10 to 15".

As noted above, one of the features of a swing motion truck is that while it may be quite stiff vertically, and while it may be resistant to parallelogram deformation because of the unsprung lateral connection member, namely the transom, frame brace, or lateral reinforcement rods, it may at the same time tend to be laterally relatively soft. A spring plank, or transom, is not necessarily the only way to prevent the truck from deforming in a parallelogram manner. Another approach is to use diagonal frame bracing between the lower portions of the side frames. The inventor herein proposes that it may be advantageous to employ a four-cornered damper arrangement, to permit the dampers to act on relatively widely spaced apart sections of the vertical side frame columns, such that they may be able to develop a significant restoring moment when the truck is deformed, thereby tending to return the truck to a square condition. Thus there may be ways to encourage maintenance of squareness in the truck while still permitting a relatively soft lateral response.

Self steering is desirable since it may tend to reduce drag and may tend to reduce wear to both the wheels and the track, and may give a smoother overall ride. One way to obtain a measure of passive self steering is to mount elastomeric pads between the pedestal seat and the bearing adapter. That is to say, when a conventional truck enters a curve, the leading wheel on the outside rail may tend to want to climb the outside rail, at least to the extent of bringing a larger radius portion of the wheel tread into contact with the outer rail. At the same time, a smaller radius portion of the tread of the inner wheel may tend to engage the inside rail. As such, the outer wheel may tend to want to pull ahead relative to the inner wheel, and the inner wheel may then tend to want to slip, or skid, somewhat. This tendency may be reduced somewhat if the leading axle is able to steer a bit relative to the trailing axle, and thereby to conform to some extent to the curve. On track having a curvature of 13 degrees per 100 foot arc, the magnitude of deflection of the axle might be of the order of 3/16" to ½" on either side (i.e., +/- ¼" in either direction = ½" total). If the axle is sprung, or otherwise biased against the steering deflection, the actual deflection may be somewhat less than this, but even a modest amount of self steering may tend to make the wheels run better through the corners, and thus may tend to reduce wear. Elastomeric pads, sometimes manufactured by Lord Corp., have sometimes been used for this purpose, and provide a resilient means for permitting some self steering to take place. However, an elastomeric pad may tend to be isotropically elastic, and thus may also tend to be prone to lateral defection such as may tend to reduce the hunting threshold on tangent (i.e., straight) track, where only longitudinal deflection is desired.

Another feature of the elastomeric pads is that their resistance to deflection is like a spring constant: it may tend to be constant without regard to the vertical load placed upon the pad. That is to say, for small deflections, the resilience of the pads may tend to exhibit the same linear force-deflection characteristic, whether the rail road car is a "light car" (i.e., empty), or fully laded. It may then tend to have roughly the same stiffness in lateral translation (i.e., lateral resistance to shear), as it has to longitudinal translation (i.e., longitudinal resistance to shear), the resistance increasing as displacement increases in a roughly linear relationship for small displacements.

However, it may be desirable to provide an anisotropic response at the interface between the bearing adapter and the pedestal seat, such that longitudinal deflection might be permitted to provide a measure of self steering, while lateral translation at the pedestal seat to bearing adapter interface is discouraged, without discouraging the lateral rocking of the sideframe more generally.

The present inventor has noted that the rocking interface surface of the bearing adapter might have a crown, or a concave curvature, like a swing motion truck, by which a rolling contact on the rocker permits lateral swinging of the side frame. The present inventor has also noted, as shown and described herein, that the bearing adapter to pedestal seat interface might also have a fore-and-aft curvature, whether a crown or a depression, and that, if used as described by the inventor hereinbelow, this crown or depression might tend to present a more or less linear resistance to deflection in the longitudinal direction, much as a spring or elastomeric pad might do. The present inventor also notes that it is generally advantageous for the rockers to be self centering.

As with the rocking of the sideframe in the lateral direction, the resistance to deflection of a crown formed in the longitudinal direction will be a function of the arcuate geometry of the interface. As noted above, in general, the larger the radius of curvature of the rocker, the shorter is its equivalent pendulum length, and the stiffer the response. In proposing a bearing adapter to pedestal seat interface that has both longitudinal and lateral arcs, the present inventor has noted that the radius of curvature of the arcs need not be the same in the longitudinal and lateral directions, thus permitting tailoring of the lateral and vertical stiffness to different needs, such as may be advantageous.

The present inventor has made two further observations. First, it has been observed in simulations that if the lateral shear constant between the pedestal seat and the bearing

adapter is the same as the longitudinal shear stiffness constant for self-steering, the hunting threshold speed may be lower than for a truck having no elastomeric pads. By contrast, if an elastomeric pad is simulated with the longitudinal shear constant as before, but with the lateral constant made infinitely large, the hunting threshold may tend to remain more or less as before.

Considering the interface between the pedestal seat and the bearing adapter, there are, potentially, 6 degrees of freedom, namely vertical, longitudinal and transverse translation, and rotation about each of the vertical, longitudinal, and lateral axes. For the purposes of analysis, in the vertical direction the connection can be approximated as being nearly infinitely stiff. In the longitudinal direction, the stiffness with an elastomeric pad is a function of the shear modulus of the elastomer, the area of the elastomer in plan view, and the thickness of the elastomer, and may be chosen to give a desired spring rate, such as, for example, 30,000 Lbs./inch in shear. Thus, for a ¼" deflection, the shear force in the pad may be about 7500 lbs. If the elastomer is isotropic, the lateral stiffness will be roughly the same. The rotational stiffness of the pad about the vertical axis may be disregarded because the deflections are very small. That is, at a 13 degree curve (i.e., 13 degrees of curvature over a 100 ft. arc length) for a standard gauge track, and assuming that steering is shared between axles, the resultant angular deflection of one axle relative to the side frames may be of the order of about 0.8 degrees (+/-). The moment generated by this deflection is most probably of the order of about 2000 in-lbs or less, or perhaps something on the order of roughly 30 – 50 lbs taken on a moment arm of the gauge width. By comparison to the longitudinal shear, this is a relatively small amount. Thus, for the purposes of analysis, the torsional resistance constant for angular deflection about the vertical axis can be taken as being near zero. By contrast, the torsional stiffness about an axis parallel to the longitudinal axis (i.e., resistance to rolling) of the sideframe may tend to be high enough to be approximated as near infinite. The torsional stiffness about the lateral axis (i.e., resistance to pitching), may also be considered very large.

However, for surfaces in rolling contact on a compound curved surface (i.e., having curvatures in two directions) the vertical stiffness may again be approximated as infinite, the longitudinal stiffness in translation at the point of contact can also be taken as infinite, the assumption being that the surfaces do not slip, and the lateral stiffness in translation at the point of contact can be taken as infinite, again, provided the surfaces do not slip. The rotational stiffness about the vertical axis may be taken as zero, since no torsion can be transmitted across the interface of a complex curvature in contact with its running surface.

For example, a ball bearing in point contact with a flat surface cannot transmit or receive, through the point of contact, torsion about an axis perpendicular to flat running surface.

By contrast, the angular stiffnesses about the longitudinal and transverse axes are non-trivial. The lateral angular stiffness gives the intended equivalent pendulum stiffnesses for the sideframe more generally. A conceptual difference between the elastomeric pad and the rolling contact approach is that sideways rocking of the sideframe in the lateral pendulum occurs at both the fore and aft pedestal seats – subject to the torsional stiffness of the sideframe itself, it can hardly do otherwise. However, as may be important in determining the hunting threshold, the lateral shear displacement of the fore and aft elastomeric pads can occur independently – one may deflect outboard, while the other deflects inboard by a different amount. Put conceptually differently, the present inventor is of the view that whereas using an elastomeric pad to obtain a measure of self steering may lower the hunting threshold, using a rolling contact device to obtain self steering may yield a desired measure of self-steering, without necessarily reducing the hunting threshold speed very much, if at all.

Second, in the past, axially aligned cylindrical rockers sharing a common longitudinal axis have been employed to increase warp stiffness by compelling the fore and aft bearing adapter interfaces to rock together. That is, in a conventional bearing adapter, the torsional stiffness in the roll direction (i.e., about the longitudinal axis of the sideframe) from the center line of the shaft of the wheelset to the crown of the bearing adapter may tend to be high enough to be approximated as being infinite for the purposes of analytical calculation. Where rockers of relatively close radii are used, (that is, where the radius of curvature of the rocker is relatively close to the radius of curvature of the seat) as for example in US Patent 5,544,591 of Armand Taillon, issued August 13, 1996, the torsional stiffness about the vertical, or z, axis of the interface between the bearing adapter crown and the pedestal seat roof may also be very high, such that it may tend to provide resistance to unsquaring relative movement between the wheelsets and side frames. However, where a complex, two dimensional, curvature is used, and the smallest female radius of curvature is larger than the largest male radius of curvature, the torsional stiffness across the bearing adapter crown to pedestal seat roof interface may be taken as being zero, as noted above.

Another observation of the present inventor is that it is desirable for the rockers to remain in rolling (i.e., static) contact, as opposed to breaking free and sliding, with resultant undesirable kinematic friction. Where torsion about the vertical axis is transmitted through a rolling surface having a horizontal axis, the ends of the rolling surface, at which the torsional moment may tend to be most aggressively transferred, (i.e., they are at the largest radial

distance from the axis of torsion) may tend to want to scuff the underlying bearing surface on which the rocker works. Scuffing of the rolling surface may tend to be undesirable, as it may lead to increased wear and a reduction in the service life of the rocker and seat.

Where a car already employs an elastomeric pad, it may be desirable also to employ a fore-and-aft rocker to obtain an additional measure of self steering without unduly softening the lateral response of the bearing adapter to pedestal seat interface. In this instance, while a two dimensional curvature may be used, the relatively low resistance of the pad to torsion about a vertical axis may permit the use of a cylindrical rocker with a transverse axis, as the elastomer may tend to provide a degree of compliance to torsion and thus may tend to reduce the tendency of a cylindrical rocker to scuff the mating surface. Alternatively, depending on the properties and performance of the elastomeric pad, it may be desirable to employ a laterally swinging rocker in addition to the elastomeric pad, such that a measure of self steering may be achieved with a side frame that rocks in the manner of a swing motion truck.

Further still, in the view of the present inventor it may be desired to increase the bearing area of the normal forces in the bearing adapter interface. In that regard, it may be advantageous to use a pair of cylindrical rockers, oriented roughly perpendicularly relative to each other, advantageously with a third element that is compliant to torsion about a vertical axis, whether that third element is an elastomeric element, or a pivot, or other element permitting at least a measure of relative rotational movement between the two cylindrical elements.

Summary of the Invention

The present invention, in its various aspects, provides a rail road car truck with bi-directional rocking at the sideframe pedestal to wheelset axle end interface. It may also provide a truck that has self steering that is proportional to the weight carried by the truck. It may further have a longitudinal rocker at the sideframe to axle end interface. Further it may provide a swing motion truck with self steering. It may also provide a swing motion truck that has the combination of a swing motion lateral rocker and an elastomeric bearing adapter pad.

In an aspect of the invention there is a sideframe pedestal to axle bearing interface assembly for a three piece rail road car truck, the interface assembly having fittings operable to rock both laterally and longitudinally.

In an additional feature of that aspect of the invention the assembly includes mating surfaces of compound curvature, the compound curvature including curvature in both lateral and horizontal directions. In another feature, the assembly includes at least one rocker element and a mating element, the rocker and mating elements being in point contact with a mating element, the element in point contact being movable in rolling point contact with the mating element. In still another feature, the element in point contact is movable in rolling point contact with the mating element both laterally and longitudinally. In yet another feature, the fittings include rockingly matable saddle surfaces.

In another feature, the fittings include a male surface having a first compound curvature and a mating female surface having a second compound curvature in rocking engagement with each other, and one of the surfaces includes at least a spherical portion. In a further feature, the fittings include a non-rocking central portion in at least one direction. In still another feature, relative to a vertical axis of rotation, rocking motion of the fittings longitudinally is torsionally de-coupled from rocking of the fittings laterally. In a yet further feature the fittings include a force transfer interface that is torsionally compliant relative to torsional moments about a vertical axis. In still another feature, the assembly includes an elastomeric member.

In another aspect of the invention, there is a swing motion three piece rail road car truck having a laterally extending truck bolster, a pair of longitudinally extending sideframes to which the truck bolster is resiliently mounted, and wheelsets to which the side frames are mounted. Damper groups are mounted between the bolster and each of the sideframes. The damper groups each have a four-cornered damper layout, and wheelset to sideframe pedestal interface assemblies operable to permit lateral swinging motion of the sideframes and longitudinal self-steering of the wheelsets.

In another aspect of the invention, there is a three piece rail road car truck having a truck bolster mounted between sideframes, and wheelsets to which the sideframes are mounted, and wheelset to sideframe interface assemblies by which to mount the sideframes to the wheelsets. The sideframe to wheelset interface assemblies include rocking apparatus to permit the sideframes to swing laterally. The rocking apparatus includes first and second surfaces in rocking engagement. At least a portion of the first surface has a first radius of curvature of less than 30 inches. The sideframe to wheelset interface includes self steering apparatus.

In a feature of that aspect of the invention, the self steering apparatus has a substantially linear force deflection characteristic. In another feature, the self steering apparatus has a force-deflection characteristic that varies with vertical loading of the sideframe to wheelset interface assembly. In a further feature, the force-deflection characteristic varies linearly with vertical loading of the sideframe to wheelset interface assembly. In another feature, the self steering apparatus includes a rocking element. In still another feature, the rocking element includes a rocking member subject to angular displacement about an axis transverse to one of the sideframes.

In another feature, the self steering apparatus includes male and female rocking elements, and at least a portion of the male rocking element has a radius of curvature of less than 40 inches. In still another feature, the self steering apparatus includes male and female rocking elements, and at least a portion of the female rocking element has a radius of curvature of less than 60 inches. In still another feature the self steering apparatus is self centering. In a further feature, the self steering apparatus is biased toward a central position.

In yet another feature, the self steering apparatus includes a resilient member. In a further feature of that further feature, the resilient member includes an elastomeric element. In another further feature, the resilient member is an elastomeric adapter pad assembly. In another feature, the resilient member is an elastomeric adapter assembly having a lateral force-displacement characteristic and a longitudinal force-displacement characteristic, and the longitudinal force-displacement characteristic is different from the lateral force-displacement characteristic. In another feature, the elastomeric adapter assembly is stiffer in lateral shear then in longitudinal shear. In again another feature, a rocker element is mounted above the elastomeric adapter pad assembly. In another feature, a rocker element is mounted directly upon the elastomeric adapter pad assembly. In a still further feature, the elastomeric adapter pad assembly includes and integral rocker member. In another feature, the three piece truck is a swing motion truck and the self steering apparatus includes an elastomeric bearing adapter pad.

In still another feature, the wheelsets have axles, and the axles have axes of rotation, and ends mounted beneath the sideframes, and, at one end of one of the axles, the self steering apparatus has a force deflection characteristic of at least one of the characteristics chosen from the set of force-deflection characteristic consisting of

(a) a linear characteristic between 3000 lbs per inch and 10,000 pounds per inch of longitudinal deflection measured at the axis of rotation at the end of the axle when the self steering apparatus bears one quarter of a vertical load of between 45,000 and 70,000 lbs.;

(b) a linear characteristic between 16,000 lbs per inch and 40,000 pounds per inch of longitudinal deflection measured at the axis of rotation at the end of the axle when the self steering apparatus bears one quarter of a vertical load of between 263,000 and 315,000 lbs.; and

(c) a linear characteristic between 0.20 and 0.75 lbs per inch of longitudinal deflection measured at the axis of rotation at the end of the axle per pound of vertical load passed into the one end of the one axle.

In another aspect of the invention there is a three piece rail road freight car truck having self steering apparatus, wherein the passive steering apparatus includes at least one longitudinal rocker.

In yet another aspect of the invention, there is a three piece rail road freight car truck having passive self steering apparatus, the self steering apparatus having a linear force-deflection characteristic, and the force-deflection characteristic varying as a function of vertical loading of the truck.

In an additional feature of that aspect of the invention, the force-displacement characteristic varies linearly with vertical loading of the truck. In another feature, the self steering apparatus includes a rocker mechanism. In another feature, the rocker mechanism is displaceable from a minimum energy state under drag force applied to a wheel of one of the wheelsets. In still another feature, the force-deflection characteristic lies in the range of between about 0.2 lbs and 0.75 lbs per inch of deflection measured at a center of and end of an axle of a wheelset of the truck per pound of vertical load passed into the end of the axle of the wheelset. In a further feature, the force deflection characteristic lies in the range of 0.27 to 0.41 lbs per inch per pound of vertical load passed into the end of the axle of the wheelset.

In yet another aspect of the invention there is a three piece rail road freight car truck having a transversely extending truck bolster, a pair of side frames mounted at opposite ends of the truck bolster, and resiliently connected thereto, and wheelsets. The sideframes are mounted to the wheelsets at sideframe to wheelset interface assemblies. At least one of the sideframe to wheelset interface assemblies is mounted between a first end of an axle of one of the wheelsets, and a first pedestal of a first of the sideframes. The wheelset to sideframe

interface assembly includes a first line contact rocker apparatus operable to permit lateral swinging of the first sideframe and a second line contact rocker apparatus operable to permit longitudinal displacement of the first end of the axle relative to the first sideframe.

In a feature of that aspect of the invention, the first and second rocker apparatus are mounted in series with a torsionally compliant member, the torsionally complaint member being compliant to torsional moments applied about a vertical axis. In another feature, a torsionally compliant member is mounted between the first and second rocker apparatus, the torsionally compliant member being torsionally compliant about a vertical axis.

In a further aspect of the invention, there is a bearing adapter for a three piece rail road freight car truck, the bearing adapter having a rocking contact surface for rocking engagement with a mating surface of a sideframe pedestal fitting, the rocking contact surface of the bearing adapter having a compound curvature.

In another feature of that aspect of the invention, the compound curvature is formed on a first male radius of curvature and a second male radius of curvature oriented cross-wise thereto. In another feature, the compound curvature is saddle shaped. In a further feature, the compound curvature is ellipsoidal. In another feature, the compound curvature is spherical.

In a still further aspect of the invention there is a three piece railroad car truck having a laterally extending truck bolster. The truck bolster has first and second ends. First and second longitudinally extending sideframes are resiliently mounted at the first and second ends of the bolster respectively. The side frames are mounted on wheelsets at sideframe to wheelset mounting interface assemblies. A four cornered damper group is mounted between each end of the truck bolster and the respective side frame to which that end is mounted. The sideframe to wheelset mounting interface assemblies are torsionally compliant about a vertical axis.

In a feature of that aspect of the invention, the truck is free of unsprung lateral cross-members between the sideframes. In another feature, the sideframes are mounted to swing laterally. In still another feature, the sideframe to wheelset mounting interface assemblies include self steering apparatus.

In another aspect of the invention there is a sideframe pedestal to axle bearing interface assembly for a three piece rail road car truck. The interface assembly has fittings operable to rock both laterally and longitudinally, and the interface assembly includes a bearing assembly having one of the rocking surface fittings defined integrally thereon.

In an additional feature of that aspect of the invention the bearing assembly includes a rocking surface of compound curvature. In another feature, the fittings include rockingly matable saddle surfaces. In yet another feature, the fittings include a male surface having a first compound curvature and a mating female surface having a second compound curvature in rocking engagement with each other. One of the surfaces includes at least a spherical portion. In still another feature, relative to a vertical axis of rotation, rocking motion of the fittings longitudinally is torsionally de-coupled from rocking of the fittings laterally. In still yet another feature, the fittings include a force transfer interface that is torsionally compliant relative to torsional moments about a vertical axis. In a further feature, the assembly includes a resilient biasing member.

In an aspect of the invention there is a sideframe pedestal to axle bearing interface assembly for a three piece rail road car truck. The interface assembly has fittings operable to rock both laterally and longitudinally, and the interface assembly includes a bearing assembly having one of the rocking surface fittings defined integrally thereon.

In an additional feature of that aspect of the invention, the bearing assembly includes a rocking surface of compound curvature. In another feature, the fittings include rockingly matable saddle surfaces. In still another feature, the fittings include a male surface having a first compound curvature and a mating female surface having a second compound curvature in rocking engagement with each other, and one of the surfaces includes at least a spherical portion. In yet another feature, relative to a vertical axis of rotation, rocking motion of the fittings longitudinally is torsionally de-coupled from rocking of the fittings laterally. In still yet another feature, the fittings include a force transfer interface that is torsionally compliant relative to torsional moments about a vertical axis. In a further feature, the assembly includes a resilient biasing member.

In another aspect of the invention, there is a sideframe pedestal to axle bearing interface assembly for a three piece rail road car truck. The interface assembly has mating rocking surfaces. The assembly includes a bearing mounted to an end of a wheelset axle.

The bearing has an outer ring, and one of the rocking surfaces is rigidly fixed relative to the bearing.

In still another aspect of the invention, there is a bearing for mounting to one end of an axle of a wheelset of a three-piece railroad car truck. The bearing has an outer member mounted in a position to permit the end of the axle to rotate relative thereto, and the outer member has a rocking surface formed thereon for engaging a mating rolling contact surface of a pedestal seat member of a sideframe of the three piece truck.

In an additional feature of that aspect of the invention, the bearing has an axis of rotation coincident with a centerline axis of the axle and the surface has a region of minimum radial distance from the center of rotation and a positive derivative $dr/d\theta$ between the region and points angularly adjacent thereto on either side.

In another feature, the surface is cylindrical. In yet another feature, the surface has a constant radius of curvature. In still another feature, the cylinder has an axis parallel to the axis of rotation of the bearing. In still yet another feature, when installed in the three piece truck, the surface has a local minimum potential energy position, the position of minimum potential energy being located between positions of greater potential energy. In yet another feature, the surface is a surface of compound curvature. In still yet another feature, the surface has the form of a saddle. In a further feature, the surface has a radius of curvature. The bearing has an axis of rotation, and a region of minimum radial distance from the axis of rotation. The radius of curvature is greater than the minimum radial distance.

In yet a further feature, there is a combination of a bearing and a pedestal seat. In an additional feature, the bearing has an axis of rotation. A first location on the surface of the bearing lies radially closer to the axis of rotation than any other location thereon; a first distance, **L** is defined between the axis of rotation and the first location. The surface of the bearing and the surface of the pedestal seat each have a radius of curvature and mate in a male and female relationship. One radius of curvature is a male radius of curvature $\mathbf{r_1}$. The other radius of curvature is a female radius of curvature, $\mathbf{R_2}$; $\mathbf{r_1}$ being greater than **L**, $\mathbf{R_2}$ is greater than $\mathbf{r_1}$, and **L**, $\mathbf{r_1}$ and $\mathbf{R_2}$ conform to the formula $\mathbf{L} - (\mathbf{r_1}^{-1} - \mathbf{R_2}^{-1}) > 0$. In another additional feature, the rocking surfaces are co-operable to permit self steering.

These and other aspects and features of the invention may be understood with reference to the detailed descriptions of the invention and the accompanying illustrations as set forth below.

Brief Description of the Figures

The principles of the invention may better be understood with reference to the accompanying figures provided by way of illustration of an exemplary embodiment, or embodiments, incorporating principles and aspects of the present invention, and in which:

Figure **1a** shows an isometric view of a railroad car truck according to the present invention;

Figure **1b** shows a side view of the railroad car truck of Figure **1a**;

Figure **1c** shows a top view of the railroad car truck of Figure **1a**;

Figure **1d** is a split view showing, in one half an end view of the truck of Figure **1a**, and in the other half and a section taken level with the truck center;

Figure **1e** shows a spring layout for the truck of Figure**1a**;

Figure **1f** shows an isometric view of an alternate embodiment of railroad car truck to that of Figure **1a**;

Figure **1g** shows a top view of the railroad car truck of Figure **1f**;

Figure **1h** shows a side view of the railroad car truck of Figure **1f**;

Figure **1i** shows an exploded view of one quarter of the truck of Figure **1f**;

Figure **2a** shows an enlarged detail of the side view of the truck of either Figure **1b** or **1g** taken at the sideframe pedestal to bearing adapter interface;

Figure **2b** shows a lateral cross-section through the sideframe pedestal to bearing adapter interface of Figure **2a**, taken at the wheelset axle centreline;

Figure **2c** shows the cross-section of Figure **2a** in a laterally deflected condition;

Figure **2d** shows a longitudinal cross-section of the pedestal seat to bearing adapter interface of Figure **2a**, taken through the longitudinal plane of symmetry of the bearing adapter;

Figure **2e** shows the longitudinal cross-section of Figure **2d** in a longitudinally deflected condition;

Figure **2f** shows a top view of the detail of Figure **2a**;

Figure **2g** shows a staggered section of the bearing adapter of Figure **2a**, on section lines '**2g – 2g**' of Figure **2a**;

Figure **3a** shows a top view of an embodiment of bearing adapter and pedestal seat such as could be used in a side frame pedestal similar to that of Figure 2a, with the seat inverted to reveal a female depression formed therein for engagement with the bearing adapter;

Figure **3b** shows a side view of the bearing adapter and seat of Figure **3a**;

Figure **3c** shows a longitudinal section of the bearing adapter of Figure **3a** taken on section '**3c** – **3c**';

Figure **3d** shows an end view of the bearing adapter and pedestal seat of Figure **3a**;

Figure **3e** shows a transverse section of the bearing adapter of Figure **3a**, taken on the wheelset axle centreline;

Figure **3f** shows a progression of longitudinal sectional profiles for the bearing adapter and seat of Figure **3a**;

Figure **3g** shows a progression of lateral sectional profiles for the bearing adapter and seat of Figure **3a**;

Figure **3h** shows a cross-section in the transverse plane of symmetry of a bearing adapter and pedestal seat pair similar to that of Figure **3e**, but having the rocker and seat portions inverted;

Figure **3i** shows a cross-section on the longitudinal plane of symmetry of the bearing adapter and pedestal seat pair of Figure **3h**;

Figure **4a** shows an isometric view of an alternate embodiment of bearing adapter and pedestal seat to that of Figure **3a** having a fully curved upper surface;

Figure **4b** shows a side view of the bearing adapter and seat of Figure **4a**;

Figure **4c** shows an end view of the bearing adapter and seat of Figure **4a**;

Figure **4d** shows a cross-section of the bearing adapter and pedestal seat of Figure **4a** taken on the longitudinal axis of symmetry;

Figure **4e** shows a cross-section of the bearing adapter and pedestal seat of Figure **4a** taken on the transverse plane of symmetry;

Figure **5a** shows a top view of an alternate bearing adapter and, an inverted view of an alternate female pedestal seat to that of Figure **3a**;

Figure **5b** shows a longitudinal section of the bearing adapter of Figure **5a**;

Figure **5c** shows an end view of the bearing adapter and seat of Figure **5a**;

Figure **6a** shows an isometric view of a further embodiment of bearing adapter and seat combination to that of Figure **3a**, in which the bearing adapter and pedestal seat have saddle shaped engagement interfaces;

Figure **6b** shows an end view of the bearing adapter and pedestal seat of Figure **6a**;

Figure **6c** shows a side view of the bearing adapter and pedestal seat of Figure **6a**;

Figure **6d** shows a lateral cross-section of the bearing adapter and pedestal seat combination of Figure **6a**;

Figure **6e** shows a longitudinal cross-section of the bearing adapter and pedestal seat combination of Figure **6a**;

Figure **6f** shows a transverse cross section of a bearing adapter and pedestal seat pair having an inverted interface to that of Figure **6a**;

Figure **6g** shows a longitudinal cross section for the bearing adapter and pedestal seat pair of Figure **6f**;

Figure **6h** shows progressive longitudinal profiles for the bearing adapter and pedestal seat of Figure **6a**;

Figure **6i** shows progressive lateral profiles for the bearing adapter and pedestal seat of Figure **6a**;

Figure **7a** shows an exploded side view of a further alternate bearing adapter and seat combination to that of Figure **3a**, having a pair of cylindrical rocker elements, and a pivoted connection therebetween;

Figure **7b** shows an exploded end view of the bearing adapter and seat of Figure **7a**;

Figure **7c** shows a cross-section of the bearing adapter and seat of Figure **7a**, as assembled, taken on the longitudinal centreline thereof;

Figure **7d** shows a cross-section of the bearing adapter and seat of Figure **7a**, as assembled, taken on the transverse centreline thereof;

Figure **8a** shows an end view of an alternate version of bearing adapter and seat assembly to that of Figure **7a** employing an elastomeric intermediate member;

Figure **8b** shows a side view of the assembly of Figure **8a**;

Figure **9a** shows an exploded side view of alternate bearing adapter and seat assembly to that of Figure **3a** or **6a**, employing an elastomeric shear pad and a laterally swinging rocker;

Figure **9b** shows a transverse cross-section of the assembly of Figure **9a**, taken on the axle center line thereof;

Figure **9c** shows a sectional view of the alternate assembly of Figure **9a**, as viewed from above taken on the staggered section indicated as '**9c – 9c**';

Figure **9d** shows a cross section of the assembly of Figure **9a** taken on the longitudinal plane of symmetry of the bearing adapter;

Figure **9e** shows an end view of an alternate rocker combination employing an elastomeric pad;

Figure **9f** shows a perspective view of an alternate pad combination to the combination of Figure **13a**.

Figure **10a** shows an isometric view of a bearing adapter for use in the assembly of Figure **9a**;

Figure **10b** shows a top view of the bearing adapter of Figure **10a**;

Figure **10c** shows a longitudinal cross-section of the bearing adapter of Figure **10a**;

Figure **11a** shows an isometric view of a pad adapter for the assembly of Figure **9a**;

Figure **11b** shows a top view of the pad adapter of Figure **11a**;

Figure **11c** shows a side view of the pad adapter of Figure **11a**;

Figure **11d** shows a half cross-section of the pad adapter of Figure **11a**;

Figure **11e** shows an isometric view of a rocker for the pad adapter of Figure **11a**;

Figure **11f** shows a top view of the rocker of Figure **11a**;

Figure **11g** shows an end view of the rocker of Figure **11a**; and

Figure **12** shows 8 alternate combinations of bi-axial rocking combinations for use as an alternative to the bearing adapter and pedestal seat combination of Figure **7a**.

Figure **13a** shows an end view of an alternate arrangement of wheelset to pedestal interface assembly arrangement to that of Figure **2a**, having mating bi-directionally arcuate rocking members, one being formed integrally as an outer portion of a bearing;

Figure **13b** shows a cross-section of the assembly of Figure **13a** taken on '**13b** – **13b**' of Figure **13a**;

Figure **13c** shows a cross-section of the assembly of Figure **13a** as viewed in the direction of arrows '**13c** – **13c**' of Figure **13b**;

Figure **14a** shows an end view of an alternate assembly to that of Figure **13a** incorporating a uni-directionally fore-and-aft rocking member;

Figure **14b** shows a cross-sectional view taken on '**14b** – **14b**' of Figure **14a**;

DETAILED DESCRIPTION OF THE INVENTION

The description that follows, and the embodiments described therein, are provided by way of illustration of an example, or examples, of particular embodiments of the principles of the present invention. These examples are provided for the purposes of explanation, and not of limitation, of those principles and of the invention. In the description, like parts are marked throughout the specification and the drawings with the same respective reference numerals. The drawings are not necessarily to scale and in some instances proportions may have been exaggerated in order more clearly to depict certain features of the invention.

In terms of general orientation and directional nomenclature, for each of the rail road car trucks described herein, the longitudinal direction is defined as being coincident with the rolling direction of the rail road car, or rail road car unit, when located on tangent (that is,

straight) track. In the case of a rail road car having a center sill, the longitudinal direction is parallel to the center sill, and parallel to the side sills, if any. Unless otherwise noted, vertical, or upward and downward, are terms that use top of rail, **TOR**, as a datum. The term lateral, or laterally outboard, refers to a distance or orientation relative to the longitudinal centerline of the railroad car, or car unit. The term "longitudinally inboard", or "longitudinally outboard" is a distance taken relative to a mid-span lateral section of the car, or car unit. Pitching motion is angular motion of a railcar unit about a horizontal axis perpendicular to the longitudinal direction. Yawing is angular motion about a vertical axis. Roll is angular motion about the longitudinal axis.

This description relates to rail car trucks and truck components. Several AAR standard truck sizes are listed at page 711 in the 1997 *Car & Locomotive Cyclopedia.* As indicated, for a single unit rail car having two trucks, a "40 Ton" truck rating corresponds to a maximum gross weight on rail (GWR) of 142,000 lbs. Similarly, "50 Ton" corresponds to 177,000 lbs, "70 Ton" corresponds to 220,000 lbs, "100 Ton" corresponds to 263,000 lbs, and "125 Ton" corresponds to 315,000 lbs. In each case the load limit per truck is then half the maximum gross car weight on rail. A "110 Ton" truck is a term sometimes used for a truck having a maximum weight on rail of 286,000 lbs.

This application refers to friction dampers, and multiple friction damper systems. There are several types of damper arrangement as shown at pp. 715 - 716 of the *1997 Car and Locomotive Encyclopedia,* those pages being incorporated herein by reference. Double damper arrangements are shown and described in my co-pending US Patent application, 10 / 210,797 entitled "Rail Road Freight Car With Damped Suspension", published as US Patent Application Publication No. US 2003/0041772 A1, on March 6, 2003, and also incorporated herein by reference. Each of the arrangements of dampers shown at pp. 715 to 716 of the *1997 Car and Locomotive Encyclopedia* can be modified according to the principles of my aforesaid co-pending application for "Rail Road Freight Car With Damped Suspension" to employ a four cornered, double damper arrangement of inner and outer dampers.

General Description of Truck Features

Figures **1a** and **1f**, provide examples of trucks **20** and **22** embodying an aspect of the present invention. Trucks **20** and **22** of Figures **1a** and **1f** can be taken as being of the same, or generally similar features and similar construction, although they may differ in pendulum length, spring stiffness, wheelbase, window width and height, and damping arrangement. That is, truck **20** of Figure **1a** may tend to have a longer wheelbase (from 73 inches to 86

inches, possibly between 80 – 84 inches for truck **20**, as opposed to a wheelbase of 63 – 73 inches for truck **22**), a main spring group having a softer vertical spring constant, and a four cornered damper group that may have different primary and secondary angles on the damper wedges. While either truck may be suitable for a variety of general purpose uses, truck **20** may be optimized to be suitable for use in rail road cars for carrying relatively low density, high value lading, such as automobiles or consumer products, for example, whereas truck **22** may be optimized for carrying denser semi-finished industrial goods, such as might be carried in rail road freight cars for transporting rolls of paper, for example. The various features of the two truck types may be interchanged, and are intended to be illustrative of a wide range of truck types in which the present invention may be employed. Notwithstanding possible differences in size, generally similar features are given the same part numbers.

Trucks **20** and **22** are symmetrical about both their longitudinal and transverse centreline axes. In each case, where reference is made to a sideframe, it will be understood that the truck has first and second sideframes, first and second spring groups, and so on.

Trucks **20** and **22** have a truck bolster, identified as **24**, and sideframes, identified as **26**. Each sideframe **26** has a generally rectangular window **28** that accommodates one of the ends **30** of the bolster **24**. The upper boundary of window **28** is defined by the sideframe arch, or compression member identified as top chord member **32**, and the bottom of window **28** is defined by a tension member identified as bottom chord **34**. The fore and aft vertical sides of window **28** are defined by sideframe columns **36**.

The ends of the tension member sweep up to meet the compression member. At each of the swept-up ends of sideframe **26** there are sideframe pedestal fittings, or pedestal seats **38**. Each fitting **38** accommodates an upper fitting, which may be a rocker or a seat, as described and discussed below. This upper fitting, whichever it may be, is indicated generically as **40**. Fitting **40** engages a mating fitting **42** of the upper surface of a bearing adapter **44**. Bearing adapter **44** engages a bearing **46** mounted on one of the ends of one of the axles **48** of the truck adjacent one of the wheels **50**. A fitting **40** is located in each of the fore and aft pedestal fittings **38**, the fittings **40** being longitudinally aligned such that the sideframe can swing transversely relative to the rolling direction of the truck.

The relationship of the mating fittings **40** and **42** is described at greater length below. The relationship of these fittings determines part of the overall relationship between an end of one of the axles of one of the wheelsets and the sideframe pedestal. That is, in

determining the overall response, the degrees of freedom of the mounting of the axle end in the sideframe pedestal involves a dynamic interface across an assembly of parts, such as may be termed a wheelset to sideframe interface assembly, that may include the bearing, the bearing adapter, an elastomeric pad, if used, a rocker if used, and the pedestal seat mounted in the roof of the sideframe pedestal. Several different embodiments of this wheelset to sideframe interface assembly are described below. To the extent that the bearing has a single degree of freedom, namely rotation of the shaft about the lateral axis, analysis of the assembly can focused on the bearing to pedestal seat interface assembly, or, on the bearing adapter to pedestal seat interface assembly. For the purposes of this description, items **40** and **42** are intended generically to represent the combination of features of a bearing adapter and pedestal seat assembly defining the interface between the roof of the sideframe pedestal and the bearing adapter, and the six degrees of freedom of motion at that interface, namely vertical, longitudinal and transverse translation (i.e., translation in the z, x, and y directions) and pitching, rolling, and yawing (i.e., rotational motion about the y, x, and z axes respectively) in response to dynamic inputs. In general, this interface can be considered to be nearly infinitely stiff in vertical translation. Discussion of the inter-relationship of the other five degrees of freedom is provided at greater length below.

Continuing with the general description of the trucks, the bottom chord or tension member of sideframe **26** may have a basket plate, or lower spring seat **52** rigidly mounted to bottom chord **34**, to give a rigid orientation relative to window **28**, and to sideframe **26** in general. Although trucks **20** and **22** are free of unsprung lateral cross-bracing, whether in the nature of a transom or lateral rods, in the event that truck **20** or truck **22** is taken to represent a "swing motion" truck with a transom or other cross bracing, the lower rocker platform of spring seat **52** may be mounted on a rocker, to permit lateral rocking relative to sideframe **26**. Spring seat **52** may have retainers for engaging the springs **54** of a spring set, or spring group, **56**, whether internal bosses, or a peripheral lip for discouraging the escape of the bottom ends the of springs. The spring group, or spring set **56**, is captured between the distal end **30** of bolster **24** and spring seat **52**, being placed under compression by the weight of the rail car body and lading that bears upon bolster **24** from above.

Bolster **24** has double, inboard and outboard, bolster pockets **60, 62** on each face of the bolster at the outboard end (i.e., for a total of 8 bolster pockets per bolster, 4 at each end). Bolster pockets **60, 62** accommodate a pair of first and second, laterally inboard and laterally outboard friction damper wedges **64, 66** and **68, 70**, respectively. Each bolster pocket **60, 62** has an inclined face, or damper seat **72**, that mates with a similarly inclined hypotenuse face

74 of the damper wedge, **64, 66, 68** and **70**. Wedges **64, 66** each sit over a first, inboard corner spring **76, 78**, and wedges **68, 70** each sit over a second, outboard corner spring **80, 82**. In this four corner arrangement, each damper is individually sprung by one or another of the springs in the spring group. The static compression of the springs under the weight of the car body and lading tends to act as a spring loading to bias the damper to act along the slope of the bolster pocket to force the friction surface against the sideframe. Friction damping is provided by damping wedges **64, 66** and **68, 70** that seat in mating bolster pockets **60, 62** that have inclined damper seats **72**. The vertical sliding faces **90** of the friction damper wedges **64, 66** and **68, 70** then ride up and down on friction wear plates **92** mounted to the inwardly facing surfaces of sideframe columns **36**. Angled faces **94** of wedges **64, 66** and **68, 70** ride against the angled face **74** of seat **72**.

When a lateral perturbation is passed to wheels **50** by the rails, rigid axles **48** may tend to cause both sideframes **26** to deflect in the same direction. The reaction of sideframes **26** is to swing, rather like pendula, on the upper rockers. The weight of the pendulum and the twisted springs may then tend to urge the sideframes back to their initial position. The tendency to oscillate harmonically due to the track perturbation may tend to be damped out be the friction of the dampers on the wear plates **92**.

As compared to bolsters with single dampers, the use of spaced apart pairs of dampers **64, 68** may tend to give a larger moment arm, as indicated by dimension "**2M**", for resisting parallelogram deformation of truck **20, 22** more generally. Placement of doubled dampers in this way may tend to yield a greater restorative "squaring" force to return the truck to a square orientation than for a single damper alone, as in truck **20, 22**. That is, in parallelogram deformation, or lozenging, the differential compression of one diagonal pair of springs (e.g., inboard spring **76** and outboard spring **82** may be more pronouncedly compressed) relative to the other diagonal pair of springs (e.g., inboard spring **78** and outboard spring **80** may be less pronouncedly compressed than springs **76** and **80**) tends to yield a restorative moment couple acting on the sideframe wear plates. This moment couple tends to rotate the sideframe in a direction to square the truck, (that is, in a position in which the bolster is perpendicular, or "square", to the sideframes). As such, the dampers co-operate in acting as biased members working between the bolster and the side frames to resist parallelogram, or lozenging, deformation of the side frame relative to the truck bolster. A middle end spring **96** bears on the underside of a land **98** located intermediate bolster pockets **60** and **62**. In the view of the present inventor, it may be that an enhanced tendency to encourage squareness at the bolster to sideframe interface (i.e., through the use of four

cornered damper groups) may tend to reduce reliance on squareness at the pedestal to wheelset axle interface. This, in turn, may tend to provide an opportunity to employ a torsionally compliant (about the vertical axis) axle to pedestal interface assembly, and to permit a measure of self steering. The top ends of the central row of springs, **100**, seat under the main central portion **102** of the end of bolster **24**.

Bearing plate **92** is significantly wider than the through thickness of the sideframes more generally, as measured, for example, at the pedestals, and wider than has been conventionally common. This additional width corresponds to the additional overall damper span width measured fully across the damper pairs, plus lateral travel as noted above, typically allowing 1 ½ (+/-) inches of lateral travel of the bolster relative to the sideframe to either side of the undeflected central position. That is, rather than having the width of one coil, plus allowance for travel, plate **92** has the width of three coils, plus allowance to accommodate 1 ½ (+/-) inches of travel to either side. Bolster **24** has inboard and outboard gibbs **106, 108** respectively, that bound the lateral motion of bolster **24** relative to sideframe columns **36**. This motion allowance may advantageously be in the range of +/- 1 $^1/_8$ to 1 $^3/_4$ inches, and is most preferably in the range of 1 $^3/_{16}$ to 1 $^9/_{16}$ inches, and can be set, for example, at 1 $^1/_2$ inches or 1 $^1/_4$ inches of lateral travel to either side of a neutral, or centered, position when the sideframe is undeflected.

The lower ends of the springs of the entire spring group, identified generally as **58**, seat in the lower spring seat **52**. Lower spring seat **52** has the layout of a tray with an upturned rectangular peripheral lip. Although truck **20** employs a spring group in a 5 x 3 arrangement, and truck **22** employs a spring group in a 3 x 3 arrangement, this is intended to be generic, and to represent a range of variations. They may represent a 2 x 4 arrangement, a 3:2:3 arrangement, and may include a hydraulic snubber, or such other arrangement of springs may be appropriate for the given service for the railcar for which the truck is intended.

Damper wedges **64, 66** and **68, 70** sit over (nominally) 44 % (+/-) of the spring group i.e., 4/9 of a 3 rows x 3 columns group. There are three related variables that are subject to optimization, namely (a) the choice, and layout of the various springs, (general arrangement of rows and columns, (b) the use (or not) of outer, inner, and inner-inner coils, use of side coils, whether outer and inner, and use of snubbers) to determine not only the overall spring stiffness, but also the proportion of that stiffness carried under the dampers; and (c) the primary angle of the wedges. There are many possible damper styles and arrangements. In general, for the same proportion of vertical damping, where a higher proportion of the total

spring stiffness is mounted under the dampers, the corresponding wedges may tend to have a larger included angle (i.e., between the wedge hypotenuse and the vertical face for engaging the friction wear plates on the sideframe columns **36**). The included angle may range from around 30 – 35 degrees to perhaps as much as 60 – 65 degrees, with a more moderate range being in the range of 35 – 45 degrees, or thereabout. The specific angle may tend to be a function of the specific spring stiffnesses and spring combinations actually employed.

One way to encourage an increase in the hunting threshold is to employ a truck having a longer wheelbase, or one whose length is proportionately great relative to its width. For example, at present two axle truck wheelbases may generally range from about 5' – 3" to 6' – 0". However, the standard North American track gauge is 4' – 8 ½", giving a wheelbase to track width ratio possibly as small as 1.12. At 6' – 0" the ratio is roughly 1.27. It would be preferable to employ a wheelbase having a longer aspect ratio relative to the track gauge.

In the case of truck **20**, the size of the spring group yields an opening between the vertical columns of sideframe more than 27 ½ inches wide. Truck **20** also has a greater wheelbase length, indicated as **WB**. **WB** is advantageously greater than 73 inches, or, taken as a ratio to the track gauge width, and is also advantageously greater than 1.30 times the track gauge width. It is preferably greater than 80 inches, or more than 1.4 times the gauge width, and in one embodiment is greater than 1.5 times the track gauge width, being as great, or greater than, about 86 inches.

Rocker Description

Now considering the interface between the sideframe pedestal and the bearing adapter, the geometry and operation of an embodiment of bearing adapter and pedestal seat assembly is first illustrated in the series of views of Figures **2a** – **2g**. Bearing adapter **44** has a lower portion **112** that is formed to accommodate, and to seat upon, bearing **46**, that is itself mounted on the end of a shaft, namely an end of axle **48**. Bearing adapter **44** has an upper portion **114** that has a centrally located, upwardly protruding fitting in the nature of a male bearing adapter interface portion **116**. A mating fitting, in the nature of a female rocker seat interface portion **118** is rigidly mounted within the roof **120** of the sideframe pedestal. To that end, laterally extending lugs **122** are mounted centrally with respect to pedestal roof **120**. The upper fitting **40**, whichever type it may be, has a body that is a plate having, along its longitudinally extending, lateral margins a set of upwardly extending lugs or ears, or tangs **124** separated by a notch, that bracket, and tightly engage lugs **122**, thereby locating upper fitting **40** in position, with the back

of the plate **126** of fitting **40** abutting the flat, load transfer face of roof **120**. In this instance, upper fitting **40** is a pedestal seat fitting with a hollowed out female bearing surface, namely portion **118**.

As shown in Figure **2g**, when the sideframes are lowered over the wheel sets, the end reliefs, or channels **128** lying between corner abutments **132** seat between the respective side frame pedestal jaws **130**. With the sideframes in place, bearing adapter **44** is thus captured in position with the male and female portions **(116** and **118)** of the adapter interface in mating engagement.

As may be noted, male portion **116** has been formed to have a generally upwardly facing surface that has both a first curvature $\mathbf{r_1}$ to permit rocking in the longitudinal direction, and a second curvature $\mathbf{r_2}$ to permit rocking (i.e. swing motion of the sideframe) in the transverse direction. Similarly, in the general case, female portion **118** has a first radius of curvature $\mathbf{R_1}$ in the longitudinal direction, and a second radius of curvature $\mathbf{R_2}$ in the transverse direction. The engagement of $\mathbf{r_1}$ with $\mathbf{R_1}$ tends to permit a rocking motion in the longitudinal direction when the wheel set exhibits a tendency to drag, with rocking displacement being generally linearly proportionate to the drag since wheel drag may be proportional to weight on the wheel. That is to say, the resistance to angular deflection is proportional to weight rather than being a fixed spring constant. This may tend to yield passive self-steering in both the light car and fully laden conditions. This relationship is shown in Figures **2d** and **2e**. Figure **2d** shows the centered, or at rest, non-deflected position of the longitudinal rocking elements. Figure **2e** shows the rocking elements at their condition of maximum longitudinal deflection. Figure **2d** represents a local, minimum potential energy condition for the system. Figure **2e** represents a system in which the potential energy has been increased by virtue of the work done by drag force **F** acting longitudinally in the horizontal plane through the center of the axle and bearing, $\mathbf{C_B}$. The present inventor has applied the following approximation for this longitudinal rocking motion:

$$\mathbf{F / \delta_{long} = k_{long} = (W / L) [[(1 / L) / (1 / r_1 - 1 / R_1)] - 1]}$$

Where:

$\mathbf{k_{long}}$ is the longitudinal constant of proportionality between longitudinal force and longitudinal deflection for the rocker.

F is a unit of longitudinal force, namely of drag on the wheel

$\boldsymbol{\delta_{long}}$ is a unit of longitudinal deflection of the centreline of the axle

L is the distance from the centreline of the axle to the apex of male portion **116**.

$\mathbf{R_1}$ is the longitudinal radius of curvature of the female hollow in the pedestal seat **38**.

$\mathbf{r_1}$ is the longitudinal radius of curvature of the crown of the male portion **116** on the bearing adapter

It will be noted that $\mathbf{R_1}$ is greater than $\mathbf{r_1}$ in this relationship, and $(1 / \mathbf{L})$ is greater than $[(1 / \mathbf{r_1}) - (1 / \mathbf{R_1})]$, and, as shown in the illustrations, **L** is smaller than either $\mathbf{r_1}$ or $\mathbf{R_1}$.

The limit of travel in the longitudinal direction is reached when the end face **134** of bearing adapter **44** extending between corner abutments **132**, comes into contact with one or other of the travel limiting abutment faces **136** of jaws **130**. In the general case, the deflection can be characterized either by the angular displacement of the centreline of the axle $\boldsymbol{\theta_1}$ or by the angular displacement of the contact point of the rocker on radius $\mathbf{r_1}$, indicated as $\boldsymbol{\theta_1}$. End face **134** of bearing adapter **44** is planar, and is relieved, or inclined, at an $\boldsymbol{\alpha}$ from the vertical. As shown in Figure **2g**, abutment face **136** is has a round, cylindrical arc, with the major axis of the cylinder extending vertically. A typical maximum radius $\mathbf{R_3}$ for this surface is 34 inches. When bearing adapter **44** is fully deflected through angle $\boldsymbol{\alpha}$, end face **134** is intended to meet abutment face **136** in line contact. When this occurs, further rocking motion of the male surface against the female surface is inhibited. Thus jaws **130** constrain the arcuate deflection of bearing adapter **44** to a limited range. A typical range for $\boldsymbol{\alpha}$ might be about 3 degrees of arc. A typical maximum value of δ_{long} may be about +/- $^{3}/_{16}$" - $^{1}/_{4}$" to either side of the vertical, at rest center line.

Similarly, as shown in Figures **2b** and **2c**, in the transverse direction, the engagement of $\mathbf{r_2}$ with $\mathbf{R_2}$ may tend to permit lateral rocking motion, in the manner of a swing motion truck. Figure **2b** shows a centered, at rest, minimum potential energy position of the lateral rocking system. Figure **2c** shows the same system in a laterally deflected condition. In this instance $\boldsymbol{\delta_2}$ is roughly $(\mathbf{L_{pendulum}} - \mathbf{r_2})\text{Sin}\varphi$, where, for small angles $\text{Sin}\varphi$ is approximately equal to φ. The present inventor has applied the following approximation for this condition, for small angular deflections:

$$\mathbf{k_{pendulum}} = (\mathbf{F_2}/\boldsymbol{\delta_2}) = (\mathbf{W}/\mathbf{L_{pend.}})[[\,(1 / \mathbf{L_{pend.}}) / ((1 / \mathbf{R_{Rocker}}) - (1 / \mathbf{R_{Seat}}))] + 1\,]$$

where:

$\mathbf{k_{pendulum}}$ = the lateral stiffness of the pendulum

$\mathbf{F_2}$ = the force per unit of lateral deflection applied at the bottom spring seat

$\boldsymbol{\delta_2}$ = a unit of lateral deflection

W = the weight borne by the pendulum

$L_{pend.}$ = the length of the pendulum, being the distance from the contact surface of the bearing adapter to the bottom of the pendulum at the spring seat

$R_{Rocker} = r_2 =$ the lateral radius of curvature of the rocker surface

$R_{Seat} = R_2 =$ the lateral radius of curvature of the rocker seat

Where R_{Seat} and R_{Rocker} are of similar magnitude, and are not unduly small relative to L, the pendulum may tend to have a relatively large lateral deflection constant. It will be noted that where R_{Seat} is large as compared to L or R_{Rocker}, or both, and can be approximated as infinite (i.e., a flat surface), this formula simplifies to:

$$k_{pendulum} = (F_{lateral} / \delta_{lateral}) = (W / L_{pendulum})[(R_{curvature} / L_{pendulum}) + 1]$$

where:

$k_{pendulum}$ = the lateral stiffness of the pendulum

$F_{lateral}$ = the force per unit of lateral deflection

$\delta_{lateral}$ = a unit of lateral deflection

W = the weight borne by the pendulum

$L_{pendulum}$ = the length of the pendulum, being the vertical distance from the contact surface of the bearing adapter to the bottom spring seat

$R_{curvature}$ = the radius of curvature of the rocker surface

Following from this, if the pendulum stiffness is taken in series with the lateral spring stiffness, then the resultant overall lateral stiffness can be obtained. Using this number in the denominator, and the design weight in the numerator yields a length, effectively equivalent to a pendulum length if the entire lateral stiffness came from an equivalent pendulum according to $L_{resultant} = W / k_{lateral\ total}$

When a lateral force is applied at the centreplate of the truck bolster, a reaction force is, ultimately, provided at the meeting of the wheels with the rail. The lateral force is transmitted from the bolster into the main spring groups, and then into a lateral force in the spring seats to deflect the bottom of the pendulum. The reaction is carried to the bearing adapter, and hence into the top of the pendulum. The pendulum will then deflect until the weight on the pendulum, multiplied by the moment arm of the deflected pendulum is sufficient to balance the moment of the lateral moment couple acting on the pendulum.

It may be noted that this bearing adapter to pedestal seat interface assembly is biased by gravity acting on the pendulum toward a central, or "at rest" position, where there is a local

minimum of the potential energy in the system. The fully deflected position shown in Figure **2c** may correspond to a deflection from vertical of the order of rather less than 10 degrees (and preferably less than 5 degrees) to either side of center, the actual maximum being determined by the spacing of gibbs **106** and **108** relative to plate **104**. Although in the general case $\mathbf{R_1}$ and $\mathbf{R_2}$ may be different such that the female surface is a section of the outside of a torus, it may be convenient, and desirable, for $\mathbf{R_1}$ and $\mathbf{R_2}$ to be the same, i.e., so that the bearing surface of the female fitting is formed as a portion of a spherical surface, having neither a major nor a minor axis, but merely being formed on a spherical radius. $\mathbf{R_1}$ and $\mathbf{R_2}$ give a desirable self-centering tendency. That tendency does not need to be overly pronounced, and the pendulum action may tend to work better if the female radius is relatively large. For that reason $\mathbf{R_1}$ and $\mathbf{R_2}$ may be as large as 60 inches, or perhaps larger. They may fall within a range of 10 inches to 100 inches, or a narrower range of 15 to 60 inches. As noted, $\mathbf{L_{pendulum}}$ may be of the order of 12 to 20 inches. The length **L** from the center of the axle to apex of the surface of the bearing adapter, at the central rest position may typically be about 5 – ¾ to 6 inches (+/-), and may be in the range of 5 – 7 inches. Bearing adapters, pedestals, side frames, and bolsters are typically made from steel. The present inventor is of the view that the rolling contact surface may preferably be made of a tool steel, or a similar material.

Further, and again in the general condition, it is preferred that the smallest of $\mathbf{R_1}$ and $\mathbf{R_2}$ be equal to or larger than the largest of $\mathbf{r_1}$ and $\mathbf{r_2}$. If this condition is true, then the contact point may have little, if any, ability to transmit torsion acting about an axis perpendicular to the point of contact, so the lateral and longitudinal rocking motions may tend to be torsionally de-coupled.

Although it is possible for $\mathbf{r_1}$ and $\mathbf{r_2}$ to be the same, such that the crowned surface of the bearing adapter (or the pedestal seat, if the relationship is inverted) is a portion of a spherical surface, in the general case it is expected that $\mathbf{r_1}$ and $\mathbf{r_2}$ will be different, with $\mathbf{r_1}$ tending to be larger, if not significantly larger, than $\mathbf{r_2}$. In the event that $\mathbf{r_1}$ and $\mathbf{r_2}$, are the same, then $\mathbf{R_1}$ and $\mathbf{R_2}$ need not be. The male fitting engagement surface thus formed will tend to be a section of the surface of a torus.

It may be noted that thus far constant radii of curvature have been assumed. While it may be practical to make the mating male and female engagement surfaces with circular arcs and constant radii of curvature, alternate arcs may be considered. For example, the surfaces may be elliptic, and may be parabolic. The surfaces may have a smaller radius of curvature in a central portion to give a generally softer lateral response for low amplitude perturbations (and

possibly relatively high frequency), with a larger radius of curvature at grater lateral angular deflection to provide a stiffer response as the magnitude of deflection increases. Alternatively, in the longitudinal direction, there may be a central portion with a large radius of curvature, to yield a relatively stiff response until the moment couple tending to cause passive self steering builds up, and then a smaller radius of curvature to ease self steering once a certain threshold has been reached. The arrangement of Figure **2a** can be inverted, such that the female engagement fitting portion may be part of bearing adapter **44**, and the male fitting may be mounted to the pedestal roof **120**.

It may be noted that the embodiment of bearing adapter to pedestal seat interface described above and shown in Figures **2a** – **2g**, may tend to have very high stiffness in vertical translation, longitudinal translation, and transverse translation, to the extent that non-slip, rolling contact is maintained. To the extent that there is point contact between the compound curvature surface of the male portion and the female portion, and the smallest radius of curvature of the female portion is larger than the largest radius of curvature of the male portion, the torsional resistance to relative rotation about the vertical, or z axis may tend to be minimal, if not zero, (i.e., it is highly torsionally compliant) and, for the purposes of analysis, torsional resistance may be taken as being zero. As such, there may not tend to be a torsional moment passed through the bearing adapter interface for the purpose of squaring the truck. Rotation about the lateral and longitudinal axes of rotation, namely the x and y axes, is non-trivial, and is determined under the equations provided above.

In the preferred embodiment the lateral rocking constant for a light car may be in the range of about 48,000 to 130,000 in-lbs per radian of angular deflection of the side frame pendulum, or, 260,000 to 700,000 in-lbs per radian for a fully laded car, or more generically, about 0.95 to 2.6 in-lbs per radian per pound of weight borne by the pendulum . Alternatively, for a light (i.e., empty) car the stiffness of the pendulum may be in the range 4,200 to 15,000 lbs per inch, and 22,000 to 61,000 lbs per inch for a fully laden 100 ton truck, or, more generically, in the range of 0.06 to 0.160 lbs per inch of lateral deflection per pound weight borne by the pendulum, as measured at the bottom spring seat.

In one embodiment $\mathbf{R_1} = \mathbf{R_2} = 15$ inches, $\mathbf{r_1} = 8 - 5/8$ inches and $\mathbf{r_2} = 5$". In another embodiment, $\mathbf{R_1} = \mathbf{R_2} = 15$ inches, and $\mathbf{r_1} = 10$" and $\mathbf{r_2} = 8 - 5/8$" (+/-). The radius of curvature of the male longitudinal rocker, $\mathbf{r_1}$, may be less than 60 inches, and may preferably lie in the range of 5 to 40 inches, and more preferably in the range of 8 to 20 inches, and may be about 15 inches. $\mathbf{R_1}$ may be less than 100 inches, and may preferably be in the range of 10 to 60 inches,

more preferably in the range of 15 to 40 inches, and may preferably be in the range of 1 – ½ to 4 times the size of $\mathbf{r_1}$. The radius of curvature of the male lateral rocker, $\mathbf{r_2}$, may be less than about 25 or 30 inches, being half, or less than half, of the 60 inch crown radius of bearing adapters of trucks that might not generally be considered to be "swing motion" trucks, and preferably lies in the range of about 5 to 20 inches. More preferably, $\mathbf{r_2}$ lies in the range of about 8 to 16 inches, and may be about 10 inches. Where line contact rocking motion is used, $\mathbf{r_2}$ may be somewhat smaller than otherwise, perhaps lying in the range of 3 to 10 inches, and perhaps being about 5 inches. $\mathbf{R_2}$ may be less than 60 inches, and may be less than about 25 or 30 inches, then being less than half the 60 inch crown radius noted above. Alternatively, $\mathbf{R_2}$ may lie in the range of 6 to 40 inches, and may lie in the range of 5 to 15 inches in the case of rolling line contact. $\mathbf{R_2}$ may be between 1 ½ to 4 times as large as $\mathbf{r_2}$. In one embodiment $\mathbf{R_2}$ may be roughly twice as large as $\mathbf{r_2}$, (+/- 20 %).

Figures 3a – 3g

Figures **3a** to **3g** show and alternate bearing adapter **144** and pedestal seat **146** pair. Bearing adapter **144** is substantially the same as bearing adapter **44**, except insofar as bearing adapter **44** has a fully curved top surface **142**, whereas bearing adapter **144** has an upper surface that has a flat central portion **148** between somewhat elevated side portions **150**. The male bearing surface portion **152** is located centrally on flat central portion **148**, and extends upwardly therefrom. As with bearing adapter **44**, bearing adapter **144** has first and second radii $\mathbf{r_1}$ and $\mathbf{r_2}$, formed in the longitudinal and transverse directions respectively, such that the upwardly protruding surface so formed is a toroidal surface. Pedestal seat **146** is substantially similar to pedestal seat fitting **38**. Pedestal seat **146** has a body having an upper surface **154** that seats in planar abutment against the downwardly facing surface of pedestal roof **120**, and upwardly extending tangs **124** that engage lugs **122** as before.

Notably, while in the general sense, the female engagement fitting portion, namely the hollow depression **156** formed in the lower face of seat **146**, is formed on longitudinal and lateral radii $\mathbf{R_1}$ and $\mathbf{R_2}$, as above, when these two radii are equal a spherical surface **158** is formed, giving the circular plan view of Figure **3a**.

As the profiles of both the male and female surfaces are compound curves (i.e., with curvatures in both the **x** and **y** directions) the inventor has provided in Figures **3f** and **3g**, a series of profiles in each of the longitudinal and transverse directions, at spaced intervals as indicated in the top view accompanying Figure **3f**. These profiles are taken at the centreline, 20 %, 40 %,

60 %, 80 %, and 100 % of the distance from the centreline to the edge of the curved portion of the bearing adapter or seat, as the case may be.

Figures **3h** and **3i** serve to illustrate that the male and female surfaces may be inverted, such that the female engagement surface **160** is formed on bearing adapter **162**, and the male engagement surface **164** on seat **166**. In this case, it is a matter of terminology which part is actually the "seat", and which is the "rocker". Sometimes the seat may be assumed to be the part that has the larger radius, and which is usually thought of as being the stationary reference, while the rocker is taken to be the part with the smaller radius, that "rocks" on the stationary seat. However, this is not always so. The essence of the relationship is that there are mating parts, whether male or female, and there is relative motion between the parts, or fittings, whether the fittings are called a "seat" or a "rocker". The fittings mate at a force transfer interface. The force transfer interface moves as the parts that co-operate to define the rocking interface rock on each other, whichever part may be, nominally, the male part or the female part. One of the mating parts or surfaces, is part of the bearing adapter, and another is part of the pedestal. There may be only two mating surfaces, or, as noted below in the context of the example of Figures **7a** – **7d**, there may be more than two mating surfaces in the overall assembly defining the dynamic interface between the bearing adapter and the pedestal fitting, or pedestal seat, however it may be called.

Figures 4a – 4e

Figures **4a** – **4e** show enlarged views of bearing adapter **44** and pedestal seat fitting **38**. As can be seen, the compound curve, upwardly facing surface **142** runs fully to terminate at the end faces **134**, and the side faces **170** of bearing adapter **44**. The side faces show the circularly downwardly arched lower walls margins **172** of side faces **170** that seat about bearings **46**. In all other respects, for the purposes of this description, bearing adapter **44** can be taken as being the same as bearing adapter **144**.

Figures 5a - 5c

Figures **5a** – **5c**, show a conceptually similar bearing adapter and pedestal seat combination to that of Figures **3a** to **3g**, but rather than having the interface portions standing proud of the remainder of the bearing adapter, the male portion **174** is sunken into the top of the bearing adapter, and the surrounding surface **176** is raised up. The mating female portion **178** while retaining its hollowed out shape, stands proud of the surrounding structure of the seat to

provide a corresponding mating surface. The longitudinally extending phantom lines indicate drain ports to discourage the collection of water.

Figures 6a - 6e

It may not be necessary for both female radii $\mathbf{R_1}$ and $\mathbf{R_2}$ to be on the same fitting, or for both male radii $\mathbf{r_1}$ and $\mathbf{r_2}$ to be on the same fitting. This is illustrated by the saddle shaped fittings of Figures **6a** to **6e**. In these illustrations, a bearing adapter **180** is of substantially the same construction as bearing adapters **44** and **144**, except insofar as bearing adapter **180** has an upper surface **192** that has a male fitting in the nature of a longitudinally extending crown **182** with a laterally extending axis of rotation, for which the radius of curvature is $\mathbf{r_1}$, and a female fitting in the nature of a longitudinally extending trough **184** having a lateral radius of curvature $\mathbf{R_2}$. Similarly, pedestal fitting **186** mounted in roof **120** has a generally downwardly facing surface **194** that has a transversely extending trough **188** having a longitudinally oriented radius of curvature $\mathbf{R_1}$, for engagement with $\mathbf{r_1}$ of crown **182**, and a longitudinally running, downwardly protruding crown **190** having a transverse radius of curvature $\mathbf{r_2}$ for engagement with $\mathbf{R_2}$ of trough **184**. A progression of sectional profiles of these inter-relating curvatures at the 0%, 20%, 40%, 60%, 80% and 100% **x** and **y** locations is provided in Figures **6h** and **6i**. In this embodiment, the smallest of $\mathbf{R_1}$ and $\mathbf{R_2}$ is again equal to or larger than the largest of $\mathbf{r_1}$ and $\mathbf{r_2}$.

As noted in the context of **3a**, in one sense the saddle shaped upper surface **192** of bearing adapter **180** is both a seat and a rocker, being a seat in one direction, and a rocker in the other, as is the pedestal seat fitting. As noted above, the essence is that there are two small radii, and two large (or possibly even infinite) radii, and the surfaces form a mating pair that engage in rolling point contact in both the lateral and longitudinal directions.

It may also be noted, as shown in Figures **6f** and **6g**, the saddle surfaces can be inverted such that whereas bearing adapter **180** has $\mathbf{r_1}$ and $\mathbf{R_2}$, bearing adapter **196** has $\mathbf{r_2}$ and $\mathbf{R_1}$. Similarly, whereas pedestal fitting **186** has $\mathbf{r_2}$ and $\mathbf{R_1}$, pedestal fitting **198** has $\mathbf{r_1}$ and $\mathbf{R_2}$.

In either case, to the extent that the smallest of $\mathbf{R_1}$ and $\mathbf{R_2}$ is larger than, or equal to, the largest of $\mathbf{r_1}$ and $\mathbf{r_2}$, the mating opposed saddle surfaces, over the desired range of motion, may tend to be torsionally uncoupled as noted above in the context of bearing adapters **44** and **144**.

Figures 7a – 7f

It may be desired that the vertical forces transmitted from the pedestal roof into the bearing adapter be passed through line contact, rather than the bi-directional rolling or rocking point contact as in the assemblies of the embodiments of Figures **2a** – **2g**, **3a** – **3i**, **4a** – **4e**, **5a** – **5c**, and **6a** – **6g**. In that case, it may be advantageous to employ an embodiment of pedestal seat to bearing adapter interface assembly having line contact rocker interfaces such as represented by the example shown in Figures **7a** to **7d**. In this instance a bearing adapter **200** has a hollowed out transverse cylindrical upper surface **202**, acting as a female engagement fitting portion formed on radius $\mathbf{R_1}$. Surface **202** may be a round cylindrical section, or it may be parabolic, or other cylindrical section.

The corresponding pedestal seat fitting **204** may have a longitudinally extending female fitting, or trough, **206** having a cylindrical surface **208** formed on radius $\mathbf{r_1}$. Again, fitting **204** is cylindrical, and may, preferably, be a round cylindrical section although, alternatively, it could be parabolic, elliptic, or some other shape for producing a rocking motion.

Trapped between bearing adapter **200** and pedestal seat fitting **204** is a rocker member **210**. Rocker member **210** has a first, or lower portion **212** having a protruding male cylindrical rocker surface **214** formed on a radius $\mathbf{r_1}$ for line contact engagement of surface **202** of bearing adapter **200** formed on radius $\mathbf{R_1}$, $\mathbf{r_1}$ being smaller than $\mathbf{R_1}$, and thus permitting longitudinal rocking to obtain passive self steering. As above, the resistance to rocking, and hence to self steering, may tend to be proportional to the weight on the rocker and hence may give proportional self steering when the car is either empty or loaded. Lower portion **212** also has an upper planar surface **216** that is preferably machined to a high level of flatness. Lower portion **212** also has a centrally located, integrally formed upwardly extending cylindrical stub **218** that stands perpendicularly proud of surface **216**. A bushing **220**, which may be a press fit bushing, mounts on stub **218**.

Rocker member **200** also has an upper portion **222** that has a second protruding male cylindrical rocker surface **224** formed on a radius $\mathbf{r_2}$ for line contact engagement with the cylindrical surface **208** of trough **206**, formed on radius $\mathbf{R_2}$, thus permitting lateral rocking of sideframe **26**. Upper portion **222** has a lower face **226**, defined in a plane perpendicular to the axis of stub **218**, again machined to a high degree of flatness, for placement in opposition to surface **216**. Upper portion **222** has a centrally located blind bore **228** of a size for tight fitting engagement of bushing **220**, such that a close tolerance, pivoting connection is obtained that is largely compliant to pivotal motion about the vertical, or z, axis of upper portion **222** with

respect to lower portion **212**. That is to say, the resistance to torsional motion about the z-axis is very small, and can be taken as zero for the purposes of analysis. To aid in this, a flat shim plate **230**, preferably of a dissimilar, non-galling metal or other suitable relatively slippery (i.e., yielding a relatively low co-efficient of friction) bearing material, has a central aperture permitting it to be installed about stub **218** and bushing **220** and is placed between opposed surfaces **206** and **216** to encourage sliding rotational motion therebetween.

In this embodiment, stub **218** could be formed in upper portion **222**, and bore **218** formed in lower portion **212**, or, alternatively, bores **228** could be formed in both upper portion **212** and lower portion **222**, and a freely floating stub **218** and bushing **220** could be captured between them. It may be noted that the angular displacement about the z axis of upper portions **222** relative to lower portion **212** may be quite small – of the order of 1 degree of arc, and may tend not to be even that large overly frequently. However, the torsional compliance provided in this way may tend to act as a preventative measure to reduce the tendency, if any, for scuffing to occur at the ends of the cylindrical rocker surfaces such as might tend otherwise to occur.

Having described the rocking portions of the assembly of Figures **7a** – **7d**, there are a number of additional features that may also be noted. First, bearing adapter **200** has longitudinally extending raised lateral abutment side walls **232** to discourage lateral migration, or escape of lower portion **212**. Lower portion **212** has non-galling, relatively low co-efficient of friction side wear shim stock members **234** that are trapped between the end faces of lower portion **212** and side walls **232**. Bearing adapter **200** may also have a drain hole formed therein, possibly centrally, or placed at an angle.

Similarly, pedestal seat fitting **204** may have laterally extending depending end abutment walls **236** to discourage longitudinal migration, or escape, of upper portion **222**. In a like manner to shim stock members **234**, non-galling, relatively low co-efficient of friction end wear shim stock members **238** are mounted between the end faces of upper portion **222** and end abutment walls **236**.

Lower portion **212** has upstanding corner abutments **240** that nest, with a gap, in the chamfered corner reliefs **242** of upper portion **222**, to limit relative rotational motion between lower portion **212** and upper portion **222**, and by permitting only limited movement of the chamfered corners of shim plate **230**, to similarly limit rotational motion of shim plate **230**.

In an alternative to the foregoing embodiment, the longitudinal cylindrical trough may be formed on the bearing adapter, and the lateral cylindrical trough may be formed in the pedestal seat, with corresponding changes in the entrapped rocker element. Further, it is not necessary that the male cylindrical portions be part of the entrapped rocker element. Rather, one of those male portions may be on the bearing adapter, and one of those male portions may be on the pedestal seat, with the corresponding female portions being formed on the entrapped rocker element. In the further alternative, the rocker element may include one male element, and one female element, having the male element formed on $\mathbf{r}_1$ (or $\mathbf{r}_2$) being located on the bearing adapter, and the female element formed on $\mathbf{R}_1$ (or $\mathbf{R}_2$) being on the underside of the entrapped rocker element, and the male element formed on $\mathbf{r}_2$ (or $\mathbf{r}_1$) being formed on the upper surface of the entrapped rocker element, and the respective mating female element formed on radius $\mathbf{R}_2$ (or $\mathbf{R}_1$) being formed on the lower face of the pedestal seat. In the still further alternative, the rocker element may include one male element, and one female element, having the male element formed on $\mathbf{r}_1$ (or $\mathbf{r}_2$) being located on the pedestal seat, and the female element formed on $\mathbf{R}_1$ (or $\mathbf{R}_2$) being on the upper surface of the entrapped rocker element, and the male element formed on $\mathbf{r}_2$ (or $\mathbf{r}_1$) being formed on the lower surface of the entrapped rocker element, and the respective mating female element formed on radius $\mathbf{R}_2$ (or $\mathbf{R}_1$) being formed on the upper face of the bearing adapter. There are, in this regard, at least eight possible combinations. It is intended that the illustrations of Figures **7a** – **7d** be understood to be generically representative of all of these possible combinations, without requiring further multiplication of drawing views.

In this way the embodiment of Figures **7a** – **7d** may tend to yield line contact at the force transfer interfaces, and yet still yield rocking in both the longitudinal and lateral direction, with compliance to torsion about the vertical axis. That is, the bearing adapter to pedestal seat interface assembly permits rotation about the longitudinal axis to give lateral rocking motion of the side frame; it permits rotation about a transverse axis to give longitudinal rocking motion; it may permit compliance to torsion about the vertical axis; it discourages lateral translation, and retains high stiffness in the vertical direction.

<u>Figures **8a** and **8b**</u>

The embodiment of Figures **8a** and **8b** is substantially similar to the embodiment of Figures **7a** to **7d**. However, rather than employing a pivot connection such as the bore, stub, and bushing of Figures **7a** – **7d**, a rocker element **244** is captured between bearing adapter **200** and pedestal seat **204**. Rocker element **244** has a torsional compliance element made of a resilient material, identified as elastomeric member **246** bonded between the opposed faces of the upper **247** and lower **245** portions of rocker element **244**. Elastomeric member **246** may be

significantly thinner than elastomeric pads generally in use heretofore, perhaps of the order of 1/10 to ¼ as thick, such as to yield a measure of torsional compliance about the z axis, such that the cylindrical elements may have a reduced tendency to scuff upon each other.

Although Figures **8a** and **8b** show the laterally extending trough in bearing adapter **200**, and the longitudinal trough in pedestal seat **204**, it will be understood that the same commentary made concerning the possible alternate variations and combinations of the features of the example of Figures **7a** to **7d** also applies to the example of Figures **8a** and **8b**.

In general, while it is preferred for the torsional element to be between the two cylindrical elements in a manner tending torsionally to decouple them, the elastomeric pad need not necessarily be installed between the two cylindrical members. For example, the rocker element **244** may be solid, and an elastomeric element may also be installed beneath the top surface of bearing adapter **200**, or above the pedestal seat element, such that a torsionally compliant element is placed in series with the two rockers. This may tend to provide the desired degree of angular compliance in the connection that may tend to prevent scuffing of the rockers.

The same general commentary may be made with regard to the pivotal connection suggested above in connection with the example of Figures **7a** to **7d**. That is, the top of the bearing adapter may be pivotally mounted to the body of the bearing adapter more generally, or the pedestal seat may be pivotally mounted to the pedestal roof, such that, again, a torsionally compliant element may be placed in series with the two rockers. However, as noted above, it is preferable that the torsionally compliant element be between the two rockers, such that they may tend to be torsionally de-coupled from each other. Again, the amount of torsional compliance may be of the order of 1 degree of rotation, which is not an overly large amount.

<u>Figures **9a** to **9c**</u>

Figures **9a** to **9c** show the combination of a bearing adapter **250** with an elastomeric bearing adapter pad **252** and a rocker **254** and pedestal seat **256** to permit lateral rocking of the sideframe.

Bearing adapter **250** may be a commercially available part. Bearing adapter **250**, shown in three additional views in Figures **10a** – **10c** is substantially similar to bearing adapter 44 (or 144) to the extent of its geometric features for engaging a bearing, but differs therefrom in having a more or less conventional upper surface. Upper surface **258** may be flat, or may have a

large (roughly 60") radius crown **260**, such as might have been used for engaging a planar pedestal seat surface. Crown **260** is split into two fore-and-aft portions, with a laterally extending central flat portion between them. Abreast of the central flat portion, bearing adapter **250** has a pair of laterally proud, outwardly facing lateral lands, **262** and **264**, and, amidst those lands, lateral lugs **266** that extend further still proud beyond lands **262** and **264**.

Bearing adapter pad **252** may be a commercially available assembly such as may be manufactured by Lord Corporation of Erie Pennsylvania, or such as may be identified as Standard Car Truck Part Number SCT 5578. Bearing adapter pad **252** has a bearing adapter engagement member in the nature of a lower plate **268** whose bottom surface **270** is relieved to seat over crown **260** in non-rocking engagement. Lateral and longitudinal translation of bearing adapter pad **252** is inhibited by an array of downwardly bent securement locating lugs, or fingers, or claws, in the nature of indexing members or tangs **272**, two per side in pairs located to reach downwardly and bracket lugs **266** in close fitting engagement. The bracketing condition with respect to lugs **266** inhibits longitudinal motion between bearing adapter pad **252** and bearing adapter **250**. The laterally inside faces of tangs **272** closely oppose the laterally outwardly facing surfaces of lands **262** and **264**, tending thereby to inhibit lateral relative motion of bearing adapter pad **252** relative to bearing adapter **250**. Given that, typically, 1/8 of the weight of the rail road car body and lading is also passed through plate **268**, its vertical, lateral, and longitudinal position relative to bearing adapter **250** can be taken as fixed.

Bearing adapter pad **252** also has an upper plate, **274**, that, in the case of a retro-fit installation of rocker **254** and seat **256**, may have been used as a pedestal seat engagement member. In any case, upper plate **274** has the general shape of a longitudinally extending channel member, with a central, or back, portion, **276** and upwardly extending left and right hand leg portions **278, 280** adjoining the lateral margins of back portion **276**. Leg portions **278** may have a size and shape such as might have been suitable for mounting directly to the sideframe pedestal.

Between lower plate **268** and upper plate **274**, bearing adapter pad **252** has a bonded resilient sandwich **280** that includes a first resilient layer, indicated as lower elastomeric layer **282** mounted directly to the upper surface of lower plate **268**, an intermediate stiffener shear plate **284** bonded or molded to the upper surface of layer **282**, and an upper resilient layer, indicated as upper elastomeric layer **286** bonded atop plate **284**. The upper surface of layer **286** is bonded or molded to the lower surface of upper plate **274**. Given that the resilient layers are quite thin as compared to their length and breadth, the resultant sandwich may tend to have

comparatively high vertical stiffness, comparatively high resistance to torsion about the longitudinal (x) and lateral (y) axes, comparatively low resistance to torsion about the vertical axis (given the small angular displacements in any case), and non-trivial, roughly equal resistance to shear in the x or y directions that may be in the range of 20,000 to 40,000 lbs per inch, or more narrowly, about 30,000 lbs per inch. Bearing adapter pad **252** may tend to permit a measure of self steering to be obtained when the elastomeric elements are subjected to longitudinal shear forces.

Rocker **254** (seen in additional views **11e**, **11f** and **11g**) has a body of substantially constant cross-section, having a lower surface **290** formed to sit in substantially flat, non-rocking engagement upon the upper surface of plate **274** of bearing adapter pad **252**, and an upper surface **292** formed to define a male rocker surface. Upper surface **292** may have a continuously radius central portion **294** lying between adjacent tangential portions **296** lying at a constant slope angle. In one embodiment, the central portion may describe 4 – 6 degrees of arc to either side of a central position, preferably about 4-½ to 5 degrees in the terminology used above, this radius is "$\mathbf{r_2}$",the male radius of a lateral rocker for permitting lateral swinging motion of side frame **26**. Where a bearing adapter with a crown radius is mounted under the resilient bearing adapter pad, the radius of rocker **254** is less than the radius of the crown, preferably less than half the crown radius, and most preferably being less than 1/3 of the crown radius. It may be formed on a radius of between 5 and 20 inches, more advantageously on a radius of between 8 and 15 inches. Surface **292** could also be formed on a parabolic profile, an elliptic or hyperbolic profile, or some other profile to yield lateral rocking.

Pedestal seat **256** (seen in Figures **11a** to **11d**) has a body having a major portion **300** that is substantially rectangular in plan view. When viewed from one end in the longitudinal direction, pedestal seat **256** has a generally channel shaped cross-section, in which major portion **300** forms the back **302** and two longitudinally running legs **304**, **306** extend upwardly and laterally outwardly from the lateral margins of major portion **300**. Legs **304** and **306** have an inner, or proximal portion **308** that extends upwardly and outwardly at an angle from the lateral margins of main portion **300**, and an outer, or distal portion, or toe **310** that extends from the end of proximal portion **308** in a substantially vertical direction. The breadth between the opposed fingers of the channel section (i.e., between opposed toes **310**) corresponds to the width of the sideframe pedestal roof **312**, as shown in the cross-section of Figure **9b**, with which legs **304** and **306** sit in close fitting, bracketing engagement. Legs **304** and **306** have longitudinally centrally located cut-outs, reliefs, rebates, or indexing features, identified as notches **314**. Notches **314** seat in close fitting engagement about T-shaped lugs **316** that are welded to the

sideframe on either side of the pedestal roof. This engagement establishes the lateral and longitudinal position of pedestal seat **256** with respect to sideframe **26**.

Pedestal seat **256** also has four laterally projecting corner lugs, or abutment fittings **318**, whose longitudinally inwardly facing surfaces oppose the laterally extending end-face surfaces of the upturned legs **278** of upper plate **274** of bearing adapter pad **252**. That is, the corner abutment fittings **318** on either lateral side of pedestal seat **256** bracket the ends of the upturned legs **278** of adapter pad **252** in close fitting engagement. This relationship fixes the longitudinal position of pedestal seat **256** relative to the upper plate of bearing adapter pad **252**.

Major portion **300** of pedestal seat **256** has a downwardly facing surface **300** that is hollowed out to form a depression defining a female rocking engagement surface **302**. This surface is formed on a female radius (identified as $\mathbf{R_2}$ in concordance with terminology used herein above) that is quite substantially larger than the radius of central portion **294** of rocker **254**, such that rocker **254** and pedestal seat **256** meet in rolling line contact engagement and permit sideframe **26** to swing laterally in a lateral rocking relationship on rocker **254**. The arcuate profile of female rocking engagement surface **302** is preferably such as to encourage lateral self centering of rocker **254**, and may have a radius of curvature that varies from a central region to adjacent regions, which may be tangential planar regions. Where pedestal seat **256** and rocker **254** are provided by way of retro-fit installation above an adapter having a crown radius, the radius of curvature of the pedestal seat may tend to be less than or equal to the crown radius. The central radius of curvature $\mathbf{R_2}$ of surface **302**, or the radius of curvature generally if constant, may be in the range of 6 to 60 inches, is preferably greater than 10 inches and less than 40 inches. It is preferably between 1 – ½ to 4 times as large as the rocker radius of curvature r_2.

As noted elsewhere, the pedestal seat need not have the female rocker surface, and the rocker need not have the male rocker surface, but rather, these surfaces could be reversed, so that the male surface is on the pedestal seat, and the female surface is on the rocker.

Particularly in the context of a retro-fit installation, there may be relatively little clearance between the upturned legs **278** of upper plate **274** and legs **304, 306** of pedestal seat **256**. This distance is shown in Figure **9b** as gap '**G**', which is preferably sufficient allowance for rocking motion between the parts that rocking motion is bounded by the spacing of the truck bolster gibbs **106, 108**.

By providing the combination of a lateral rocker and a shear pad, the resultant assembly may provide an anisotropic response at the bearing adapter to pedestal seat assembly interface, with a generally increased softness in the lateral direction, while permitting a measure of self steering.

The example of Figure **9a** may be provided as an original installation, or may be provided as a retrofit installation. In the case of a retrofit installation, rocker **254** and pedestal seat **256** are installed between an existing elastomeric pad and an existing pedestal seat, or, preferably, may be installed in addition to a replacement elastomeric pad of lesser through-thickness, such that the overall height of the bearing adapter to pedestal seat interface may remain roughly the same as it was before the retrofit.

Figures **9e** and **9f** represent alternate embodiments of combinations of elastomeric pads and rockers. While the embodiment of Figure **9a** showed an elastomeric sandwich that had roughly equivalent response to shear in the lateral and longitudinal directions, this need not be the general case. For example, in the embodiments of Figures **9e** and **9f**, elastomeric bearing adapter pad assemblies **330** and **332** have respective resilient elastomeric sandwich laminates, indicated generally as **334** and **336** in which the stiffeners **326**, **350** have longitudinally extending corrugations, or waves. In the longitudinal direction, the sandwich may tend to react in nearly pure shear, as before in the example of Figure **9a**. However, deflection in the lateral direction now requires not only a shear component, but also a component normal to the elastomeric elements, in compressive or tensile stress, rather than, and in addition to, shear. This may tend to give a stiffer lateral response, and hence an anisotropic response. It will be understood that an anisotropic shear pad arrangement of this nature could have been used in the embodiment of Figure **9a**, and a planar arrangement, as in the embodiment of Figure **9a** could be used in either of the embodiments of Figures **9e**, and **9f**.

Considering Figure **9e**, both base plate **338** and upper plate **340** has a wavy contour corresponding to the wavy contour of sandwich **334** more generally. Rocker **342** has a lower surface of corresponding profile. Otherwise, this embodiment is substantially the same as the embodiment of Figure **9a**.

Considering Figure **9f**, an elastomeric bearing adapter pad assembly **332** has a base plate **344** having a lower surface for seating in non-rocking relationship on a bearing adapter, in the same manner as bearing adapter pad assembly **252** sits upon bearing adapter **250**. The upper surface **346** of base plate **344** has a corrugated, or wavy contour, the corrugations running

lengthwise, as discussed above. An elastomeric laminate of a first resilient layer **348**, an internal stiffener plate **350**, and a second resilient layer **352** is located between base plate **344** and a correspondingly wavy undersurface of upper plate **354**. Rather than being a flat plate upon which a further rocker plate is mounted, upper plate **354** has an upper surface **356** having an integrally formed rocker contour corresponding to that of the upper surface of rocker **252**. Pedestal seat **358** then mounts directly to, and in lateral rocking relationship with upper plate **354**, without need for a separate rocker part. A steel wear plate, identified as pedestal seat **358**, is intended to be substantially the same as pedestal seat **256**. The combination of bearing adapter pad **332** and pedestal seat **356** may have interconnecting abutments **362** to prevent longitudinal migration of rocker surface **356** relative to the contoured downwardly facing surface **360** of pedestal seat **356**.

Figure 12

Figure **12** shows eight different variations of alternative bearing adapter to pedestal seat interface assembly bi-directional rocker arrangements to permit lateral swing motion of the sideframe and longitudinal rocking to permit a measure of self steering. In each case, for simplicity of illustration, the variations are shown in a largely schematic form to illustrate the relationship of the various curved surfaces. It will be understood that in each case, the bearing adapter is intended to have the general form of bearing adapter **44** or **144**, and that the pedestal seat is intended to have the general form of pedestal seat fitting **204**, subject to the indicated arrangement of curved rocking engagement surfaces. In particular, these eight variations correspond to the possible variations of the example of Figure **7a**, differing therefrom in having a solid intermediate element rather than the two part intermediate rocker element of Figure **7a**.

Keeping the foregoing in mind, in the first variation, at top left, a bearing adapter **370** has a laterally extending male cylindrical upper surface **372** formed on a radius of curvature $\mathbf{r_1}$ to permit longitudinal rocking such as may permit self steering. Pedestal seat **374** has a longitudinally extending male cylindrical surface **376** formed on a radius of curvature $\mathbf{r_2}$ such as may permit lateral swing motion rocking of sideframe **26**. Intermediate rocker member **378** has a lower cylindrical female surface **380** formed on radius of curvature $\mathbf{R_1}$ for mating rocking line contact engagement with surface **372**, and an upper female cylindrical surface **382** formed on a radius of curvature $\mathbf{R_2}$ for mating rocking line contact engagement with surface **376**.

Immediately to the right of the first variation is the second variation, in which a bearing adapter **390** has a laterally extending male cylindrical upper surface **392** formed on a radius of

curvature r_1 to permit longitudinal rocking such as may permit self steering. Pedestal seat **394** has a longitudinally extending female cylindrical surface **396** formed on a radius of curvature R_2 such as may permit lateral swing motion pendulum rocking of sideframe **26**. Intermediate rocker member **398** has a lower cylindrical female surface **400** formed on radius of curvature R_1 for mating rocking line contact engagement with surface **372**, and an upper male cylindrical surface **402** formed on a radius of curvature r_2 for mating rocking line contact engagement with surface **396**.

Immediately to the right of the second variation is the third variation, in which a bearing adapter **410** has a laterally extending female cylindrical upper surface **412** formed on a radius of curvature R_1 to permit longitudinal rocking such as may permit self steering. Pedestal seat **414** has a longitudinally extending male cylindrical surface **416** formed on a radius of curvature r_2 such as may permit lateral swing motion rocking of sideframe **26**. Intermediate rocker member **418** has a lower cylindrical male surface **420** formed on radius of curvature r_1 for mating rocking line contact engagement with surface **412**, and an upper female cylindrical surface **422** formed on a radius of curvature R_2 for mating rocking line contact engagement with surface **416**.

Immediately to the right of the third variation is the fourth variation, in which a bearing adapter **430** has a laterally extending female cylindrical upper surface **432** formed on a radius of curvature R_1 to permit longitudinal rocking such as may permit self steering. Pedestal seat **434** has a longitudinally extending female cylindrical surface **436** formed on a radius of curvature R_2 such as may permit lateral swing motion rocking of sideframe **26**. Intermediate rocker member **444** has a lower cylindrical male surface **440** formed on radius of curvature r_1 for mating rocking line contact engagement with surface **432**, and an upper female cylindrical surface **442** formed on a radius of curvature r_2 for mating rocking line contact engagement with surface **436**.

Immediately below the first variation is the fifth variation, in which a bearing adapter **450** has a longitudinally extending male cylindrical upper surface **452** formed on a radius of curvature r_2 to permit lateral rocking such as may permit lateral swing motion rocking of sideframe **26**. Pedestal seat **454** has a laterally extending female cylindrical surface **456** formed on a radius of curvature R_1 such as may permit self steering. Intermediate rocker member **458** has a longitudinally extending lower cylindrical female surface **460** formed on radius of curvature R_2 for mating rocking line contact engagement with surface **452**, and a laterally extending upper male cylindrical surface **462** formed on a radius of curvature r_1 for mating rocking line contact engagement with surface **456**.

Immediately to the right of the fifth variation is the sixth variation, in which a bearing adapter **470** has a longitudinally extending male cylindrical upper surface **472** formed on a radius of curvature $\mathbf{r_2}$ to permit lateral rocking such as may permit lateral swinging motion rocking of sideframe **26**. Pedestal seat **474** has a laterally extending male cylindrical surface **476** formed on a radius of curvature $\mathbf{r_1}$ such as may permit self steering. Intermediate rocker member **478** has a longitudinally extending lower cylindrical female surface **480** formed on radius of curvature $\mathbf{R_2}$ for mating rocking line contact engagement with surface **472**, and a laterally extending upper female cylindrical surface **482** formed on a radius of curvature $\mathbf{R_1}$ for mating rocking line contact engagement with surface **476**.

Immediately to the right of the sixth variation is the seventh variation, in which a bearing adapter **490** has a longitudinally extending female cylindrical upper surface **492** formed on a radius of curvature $\mathbf{R_2}$ to permit lateral rocking such as may permit lateral swinging motion rocking of sideframe **26**. Pedestal seat **494** has a longitudinally extending female cylindrical surface **496** formed on a radius of curvature $\mathbf{R_1}$ such as may permit self steering. Intermediate rocker member **498** has a longitudinally extending lower cylindrical male surface **500** formed on radius of curvature $\mathbf{r_2}$ for mating rocking line contact engagement with surface **492**, and a laterally extending upper male cylindrical surface **502** formed on a radius of curvature $\mathbf{r_1}$ for mating rocking line contact engagement with surface **496**.

Immediately to the right of the seventh variation is the eighth variation, in which a bearing adapter **510** has a longitudinally extending female cylindrical upper surface **512** formed on a radius of curvature $\mathbf{R_2}$ to permit lateral rocking such as may permit lateral swing motion rocking of sideframe **26**. Pedestal seat **514** has a longitudinally extending male cylindrical surface **516** formed on a radius of curvature $\mathbf{r_1}$ such as may permit self steering. Intermediate rocker member **518** has a longitudinally extending lower cylindrical male surface **520** formed on radius of curvature $\mathbf{r_2}$ for mating rocking line contact engagement with surface **512**, and a laterally extending upper female cylindrical surface **522** formed on a radius of curvature $\mathbf{R_1}$ for mating rocking line contact engagement with surface **516**.

In general, provided that the radii employed yield a physically appropriate combination tending toward a stable minimum energy state, the male portion of the bearing adapter to pedestal seat interface (with the smaller radius of curvature) may be on either the bearing adapter or on the pedestal seat, and the mating female portion (with the larger radius of curvature) may be on the other part, whichever it may be. In that light, although a particular

depiction may show a male portion on a bearing adapter, and a female fitting on the pedestal seat, it is understood that these features can, in general, be reversed, without requiring a multiplicity of drawings to show all possible permutations.

Figures 13a to 13c

Rather than employ a bearing adapter that is separate from the bearing, Figures **13a** to **13c** show a bearing **520** mounted on one of the end of an axle **522**. Bearing **520** has an integrally formed arcuate rolling contact surface **524** for mating rolling point contact with a mating rolling contact surface **526** of a pedestal seat fitting **528**. The general geometry of the rolling relationship is as described above in terms of the possible relationships of $\mathbf{r_1}$, $\mathbf{R_1}$ and **L**, and, as noted above, the male and female rolling contact surfaces can be reversed, such that the male surface is on the pedestal seat, and the female surface is on the bearing, or further still, in the case of a compound curvature, the surfaces made be saddle shaped, as described above. The bearing illustrations of Figures **13b** and **14b** are based on the bearing cross-section illustration shown on page 812 of the *1997 Car and Locomotive Cyclopedia*. That illustration was provided to the *Cyclopedia* courtesy of Brenco Inc., of Petersburg, Virginia.

In greater detail, bearing **520** is an assembly of parts including an inner ring **530**, a pair of tapered roller assemblies **532** whose inner ring engages axle **522**, and an outer ring member **534** whose inner frustoconical bearing surfaces engage the rollers of assemblies **532**. The entire assembly, including seals, spacers, and backing ring is held in place by an end cap **536** mounted to the end of axle **522**. In the assembly of Figures **13a** to **13c**, does not employ a round cylindrical outer ring member, but rather, ring member **534** is made with an upper portion **540** having the same general shape and function as bearing adapter **44** or **144**, including tapered end walls **538** for rocking motion travel limiting abutment against the surfaces of the pedestal jaws **130** as described above. Further, upper portion **540** includes corner abutments **544** for bracketing jaws **130**, again, as described above. Thus a bearing is provided with an integrally formed rocking surface. The rocking surface is permanently fixed with relation to the remainder of the underlying bearing assembly. In this way, an assembly is provided in which rotation of the bearing housing is inhibited relative to the rocking surface.

In Figures **14a** and **14b**, an integrated bearing and bearing adapter rocker assembly, or wheelset to pedestal interface assembly, is indicated as modified bearing **560**. In this case the outer ring **562** has been formed in the shape of a laterally extending, cylindrical rocker surface **564**, such as a male surface (although it could be female as discussed above), for engaging the

mating female (although, as discussed, it could be male) laterally rocker surface **566** of pedestal seat **568**, such as may tend to provide weight-proportional self steering, as discussed above.

Thus, the embodiments of Figures **13a** and **14a** both show a sideframe pedestal to axle bearing interface assembly for a three piece rail road car truck. The assembly of the embodiment of Figure **13a** has fittings that are operable to rock both laterally and longitudinally. Both embodiments include bearing assemblies having one of the rocking surface fittings, whether male or female, of saddle shape, formed as an integral portion of the outer ring of the bearing, such that the location of the rolling contact surface is rigidly located relative to the bearing (because, in this instance, it is part of the bearing). In the embodiment of Figure **13a**, the integrally firmed surface is a compound surface, whereas in the embodiment of Figure **14b**, the rolling contact surface is a cylindrical surface, which may be formed on an arc of constant radius of curvature.

The possible permutations of surface types include those indicated above in terms of a two element interface (i.e., the rocking surface on the top of the bearing, and the mating rocking surface on the pedestal seat) or a three element interface, in which an intermediate rocking member is mounted between (a) the surface rigidly located with respect to the bearing races, and (b) the surface of the pedestal seat. As above, one or another of the surfaces may be formed on a spherical arc portion such that the fittings are torsionally compliant, or, put alternatively, torsionally de-coupled with respect to rotation about the vertical axis.

Each of the assemblies of Figures **13a** and **14a** has a bearing for mounting to one end of an axle of a wheelset of a three-piece railroad car truck. The bearing has an outer member mounted in a position to permit the end of the axle to rotate relative thereto, inasmuch as the inner ring is intended to rotate with respect to the outer ring. The bearing has an axis of rotation, about which its rings and bearings are concentric that, when installed, may tend to be coincident with the longitudinal axis of the axis of the axle of the wheelset. In each case, the outer member has a rocking surface formed thereon for engaging a mating rolling contact surface of a pedestal seat member of a sideframe of the three piece truck.

The rolling contact surface of the bearing has a local minimum energy condition when centered under the corresponding seat, and it is preferred that the mating rolling contact surface be given a radius that may tend to encourage self centering of the male rolling contact element. That is to say, displacement from the minimum energy position (preferably the

centered position) may tend to cause the vertical separation distance between the centerline of the wheelset axis (and hence the centreline of the axis of rotation of the bearing) to become more distantly spaced from the sideframe pedestal roof, since the rocking action may tend marginally to raise the end of the sideframe, thus increasing the stored potential energy in the system.

This can be expressed differently. In cylindrical polar co-ordinates, the long axis of the wheelset axle may be considered as the axial direction. There is a radial direction measured perpendicularly away from the axial direction, and there is an angular circumferential direction that is mutually perpendicular to both the axial direction, and the radial direction. There is a location on the rolling contact surface that is closer to the axis of rotation of the bearing than any other location. This defines the "rest" or local minimum potential energy equilibrium position. Since the radius of curvature of the rolling contact surface is greater than the radial length, **L**, between the axis of rotation of the bearing and the location of minimum radius, the radial distance, as a function of circumferential angle θ will increase to either side of the location of minimum radius (or, put alternatively, the location of minimum radial distance from the axis of rotation of the bearing lies between regions of greater radial distance). Thus the slope of the function $\mathbf{r}(\theta)$, namely $\mathbf{dr/d\theta}$, is zero at the minimum point, and is such that **r** increases at an angular displacement away from the minimum point to either side of the location of minimum potential energy. Where the surface has compound curvature, both $\mathbf{dr/d\theta}$ and $\mathbf{dr/dL}$ are zero at the minimum point, and are such that **r** increases to either side of the location of minimum energy to all sides of the location of minimum energy, and zero at that location. This may tend to be true whether the rolling contact surface on the bearing is a male surface or a female surface or a saddle, and whether the center of curvature lies below the center of rotation of the bearing, or above the rolling contact surfaces. The curvature of the rolling contact surface may be spherical, ellipsoidal, toroidal, paraboloid, parabolic or cylindrical. The rolling contact surface has a radius of curvature, or radii of curvature, if a compound curvature is employed, that is, or are, larger than the distance from the location of minimum distance from the axis of rotation, and the rolling contact surfaces are not concentric with the axis of rotation of the bearing.

Another way to express this is to note that there is a first location on the rolling contact surface of the bearing that lies radially closer to the axis of rotation of the bearing than any other location thereon. A first distance, **L** is defined between the axis of rotation, and that nearest location. The surface of the bearing and the surface of the pedestal seat each have a radius of curvature and mate in a male and female relationship, one radius of

curvature being a male radius of curvature $\mathbf{r_1}$, the other radius of curvature being a female radius of curvature, $\mathbf{R_2}$, (whichever it may be). $\mathbf{r_1}$ is greater than **L**, $\mathbf{R_2}$ is greater than $\mathbf{r_1}$, and **L**, $\mathbf{r_1}$ and $\mathbf{R_2}$ conform to the formula $\mathbf{L}^{-1} - (\mathbf{r_1}^{-1} - \mathbf{R_2}^{-1}) > 0$, the rocker surfaces being co-operable to permit self steering.

The inventor presently prefers the embodiments of Figures **4a** and **6a**. However, there are many possible combinations and permutations of the features of the examples shown herein. In general, it is thought that a self draining geometry may be preferable over one in which a hollow is formed and for which a drain hole may be required.

In each of the trucks shown and described herein, the overall ride quality may depend on the inter-relation of the spring group layout and physical properties, or the damper layout and properties, or both, in combination with the dynamic properties of the bearing adapter to pedestal seat interface assembly. The present inventor considers it to be advantageous for the lateral stiffness of the sideframe acting as a pendulum to be less than the lateral stiffness of the spring group in shear. In rail road cars having 100 ton trucks, one embodiment may employ trucks having vertical spring group stiffnesses in the range of 18,000 Lbs./inch to 36,000 Lbs./inch in combination with an embodiment of bi-directional bearing adapter to pedestal seat interface assemblies as shown and described herein. In another embodiment, the vertical stiffness of the spring group may be less than 12,000 Lbs./in per spring group, with a horizontal shear stiffness of less than 6000 Lbs./in.

In either case, the sideframe pendulum may have a vertical length measured (when undeflected) from the rolling contact interface at the upper rocker seat to the bottom spring seat of between 12 and 20 inches, preferably between 14 and 18 inches. The equivalent length $\mathbf{L_{eq}}$, may be in the range of 8 to 20 inches, depending on truck size and rocker geometry, and is preferably in the range of 11 to 15 inches, and is most preferably between about 7 and 9 inches for 28 inch wheels (70 ton "special"), between about 8 ½ and 10 inches for 33 inch wheels (70 ton), 9 ½ and 12 inches for 36 inch wheels (100 or 110 ton), and 11 and 13 ½ inches for 38 inch wheels (125 ton). Although truck **20** or **22** may be a 70 ton special, a 70 ton, 100 ton, 110 ton, or 125 ton truck, it is preferred that truck **20** or **22** be a truck size having 33 inch diameter, or even more preferably 36 or 38 inch diameter wheels.

In the trucks described herein, for their fully laden design condition which may be determined either according to the AAR limit for 70, 100, 110 or 125 ton trucks, or, where a lower intended lading is chosen, then in proportion to the vertical sprung load yielding 2

inches of vertical spring deflection in the spring groups, the equivalent lateral stiffness of the sideframe, being the ratio of force to lateral deflection measured at the bottom spring seat, is less than the horizontal shear stiffness of the springs. The equivalent lateral stiffness of the sideframe $k_{sideframe}$ is less than 6000 Lbs./in. and preferably between about 3500 and 5500 Lbs./in., and more preferably in the range of 3700 – 4100 Lbs./in. By way of an example, in one embodiment a 2 x 4 spring group has 8 inch diameter springs having a total vertical stiffness of 9600 Lbs./ in. per spring group and a corresponding lateral shear stiffness $k_{spring\ shear}$ of 4800 Lbs./in. The sideframe has a rigidly mounted lower spring seat. It is used in a truck with 36 inch wheels. In another embodiment, a 3 x 5 group of 5 ½ inch diameter springs is used, also having a vertical stiffness of about 9600 Lbs./in. in a truck with 36 inch wheels. It is intended that the vertical spring stiffness per spring group be in the range of less than 30,000 Lbs./in., that it advantageously be in the range of less than 20,000 Lbs./in and that it preferably be in the range of 4,000 to 12000 Lbs./in, and most preferably be about 6000 to 10,000 Lbs./in. The twisting of the springs has a stiffness in the range of 750 to 1200 Lbs./in. and a vertical shear stiffness in the range of 3500 to 5500 Lbs./in. with an overall sideframe stiffness in the range of 2000 to 3500 Lbs./in.

In the embodiments of trucks in which there is a fixed bottom spring seat, the truck may have a portion of stiffness, attributable to unequal compression of the springs equivalent to 600 to 1200 Lbs./in. of lateral deflection, when the lateral deflection is measured at the bottom of the spring seat on the sideframe. Preferably, this value is less than 1000 Lbs./in., and most preferably is less than 900 Lbs./in. The portion of restoring force attributable to unequal compression of the springs will tend to be greater for a light car as opposed to a fully laden car, i.e., a car laden in such a manner that the truck is approaching its nominal load limit, as set out in the 1997 *Car and Locomotive Cyclopedia* at page 711.

The double damper arrangements shown above can also be varied to include any of the four types of damper installation indicated at page 715 in the *1997 Car and Locomotive Cyclopedia*, whose information is incorporated herein by reference, with appropriate structural changes for doubled dampers, with each damper being sprung on an individual spring. That is, while inclined surface bolster pockets and inclined wedges seated on the main springs have been shown and described, the friction blocks could be in a horizontal, spring biased installation in a pocket in the bolster itself, and seated on independent springs rather than the main springs. Alternatively, it is possible to mount friction wedges in the sideframes, in either an upward orientation or a downward orientation.

The embodiments of trucks shown and described herein may vary in their suitability for different types of service. Truck performance can vary significantly based on the loading expected, the wheelbase, spring stiffnesses, spring layout, pendulum geometry, damper layout and damper geometry.

Various embodiments of the invention have now been described in detail. Since changes in, or additions to, the above-described embodiments may be made without departing from the nature, spirit or scope of the invention, the invention is not to be limited to those details but only by the appended claims.

Claims

I claim:

1. A sideframe pedestal to axle bearing interface assembly for a three piece rail road car truck, said interface assembly having fittings operable to rock both laterally and longitudinally, and said interface assembly including a bearing assembly having one of said rocking surface fittings defined integrally thereon.

2. The sideframe pedestal to axle bearing interface assembly of claim 1 wherein said bearing assembly includes a rocking surface of compound curvature.

3. The sideframe pedestal to axle bearing interface assembly of claim 1 wherein said fittings include rockingly matable saddle surfaces.

4. The sideframe pedestal to axle bearing interface assembly of claim 1 wherein said fittings include a male surface having a first compound curvature and a mating female surface having a second compound curvature in rocking engagement with each other, and one of said surfaces includes at least a spherical portion.

5. The sideframe pedestal to axle bearing interface assembly of claim 1 wherein, relative to a vertical axis of rotation, rocking motion of said fittings longitudinally is torsionally de-coupled from rocking of said fittings laterally.

6. The sideframe pedestal to axle bearing interface assembly of claim 1 wherein said fittings include a force transfer interface that is torsionally compliant relative to torsional moments about a vertical axis.

7. The sideframe pedestal to axle bearing interface assembly of claim 1 wherein said assembly includes a resilient biasing member.

8. A sideframe pedestal to axle bearing interface assembly for a three piece rail road car truck, said interface assembly having mating rocking surfaces, said assembly including a bearing mounted to an end of a wheelset axle, said bearing having an outer ring, and one of said rocking surfaces is rigidly fixed relative to said bearing.

9. A bearing for mounting to one end of an axle of a wheelset of a three-piece railroad car truck, said bearing having an outer member mounted in a position to permit the end of the axle to rotate relative thereto, and said outer member has a rocking surface formed thereon for engaging a mating rolling contact surface of a pedestal seat member of a sideframe of the three piece truck.

10. The bearing of claim 9 wherein said bearing has an axis of rotation coincident with a centerline axis of the axle and said surface has a region of minimum radial distance from said center of rotation and a positive derivative $dr/d\theta$ between said region and points angularly adjacent thereto on either side.

11. The bearing of claim 10 wherein said surface is cylindrical.

12. The bearing of claim 11 wherein said surface has a constant radius of curvature.

13. The bearing of claim 11 wherein said cylinder has an axis parallel to the axis of rotation of the bearing.

14. The bearing of claim 9 wherein, when installed in the three piece truck, said surface has a local minimum potential energy position, said position of minimum potential energy being located between positions of greater potential energy.

15. The bearing of claim 9 wherein said surface is a surface of compound curvature.

16. The bearing of claim 9 wherein said surface has the form of a saddle.

17. The bearing of claim 9 wherein said surface has a radius of curvature, said bearing has an axis of rotation, and a region of minimum radial distance from said axis of rotation, said radius of curvature being greater than said minimum radial distance.

18. A combination of the bearing of claim 9 and the pedestal seat.

19. The combination of claim 18 wherein said bearing has an axis of rotation, a first location on said surface of said bearing lies radially closer to said axis of rotation than any other location thereon; a first distance, L is defined between said axis of rotation and said first location, said surface of said bearing and said surface of said pedestal seat each have a

radius of curvature and mate in a male and female relationship, one radius of curvature being a male radius of curvature $\mathbf{r_1}$, the other radius of curvature being a female radius of curvature, $\mathbf{R_2}$; $\mathbf{r_1}$ being greater than $\mathbf{L}$, $\mathbf{R_2}$ is greater than $\mathbf{r_1}$, and $\mathbf{L}$, $\mathbf{r_1}$ and $\mathbf{R_2}$ conform to the formula $\mathbf{L}^{-1} - (\mathbf{r_1}^{-1} - \mathbf{R_2}^{-1}) > 0$.

20. The combination of claim 18 wherein said rocking surfaces are co-operable to permit self steering.

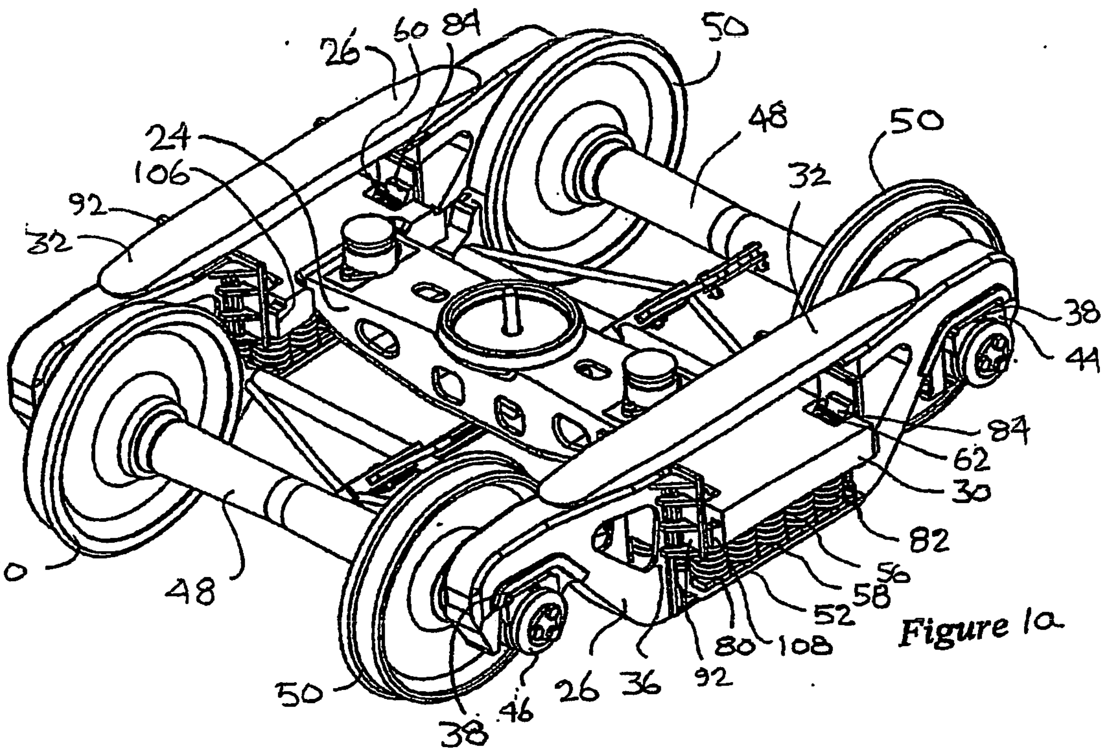

Figure 1a

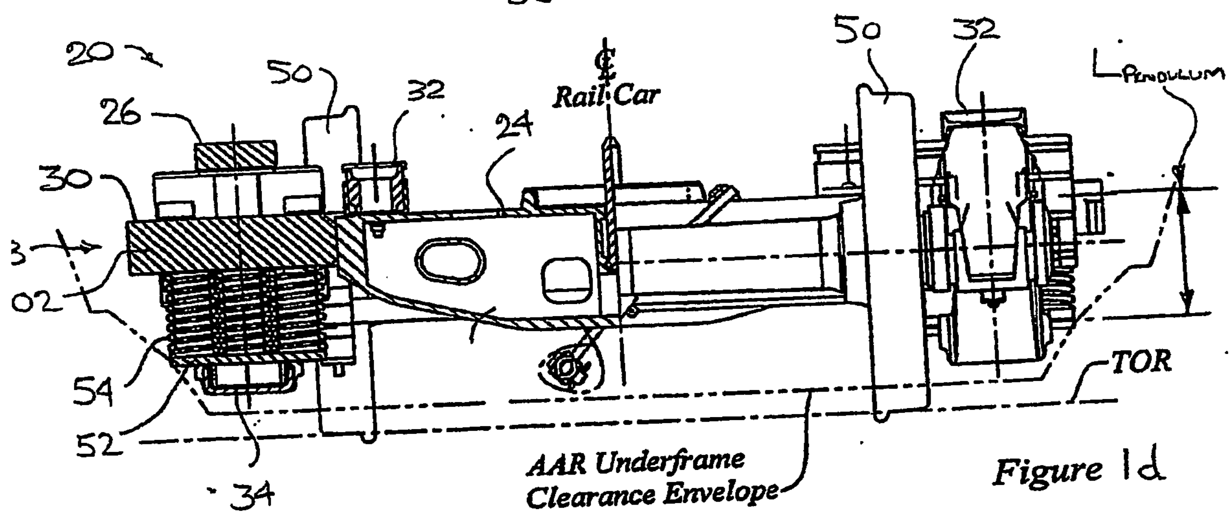

Figure 1d

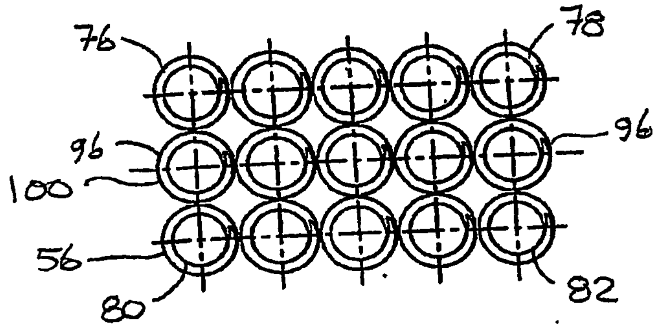

Figure 1e

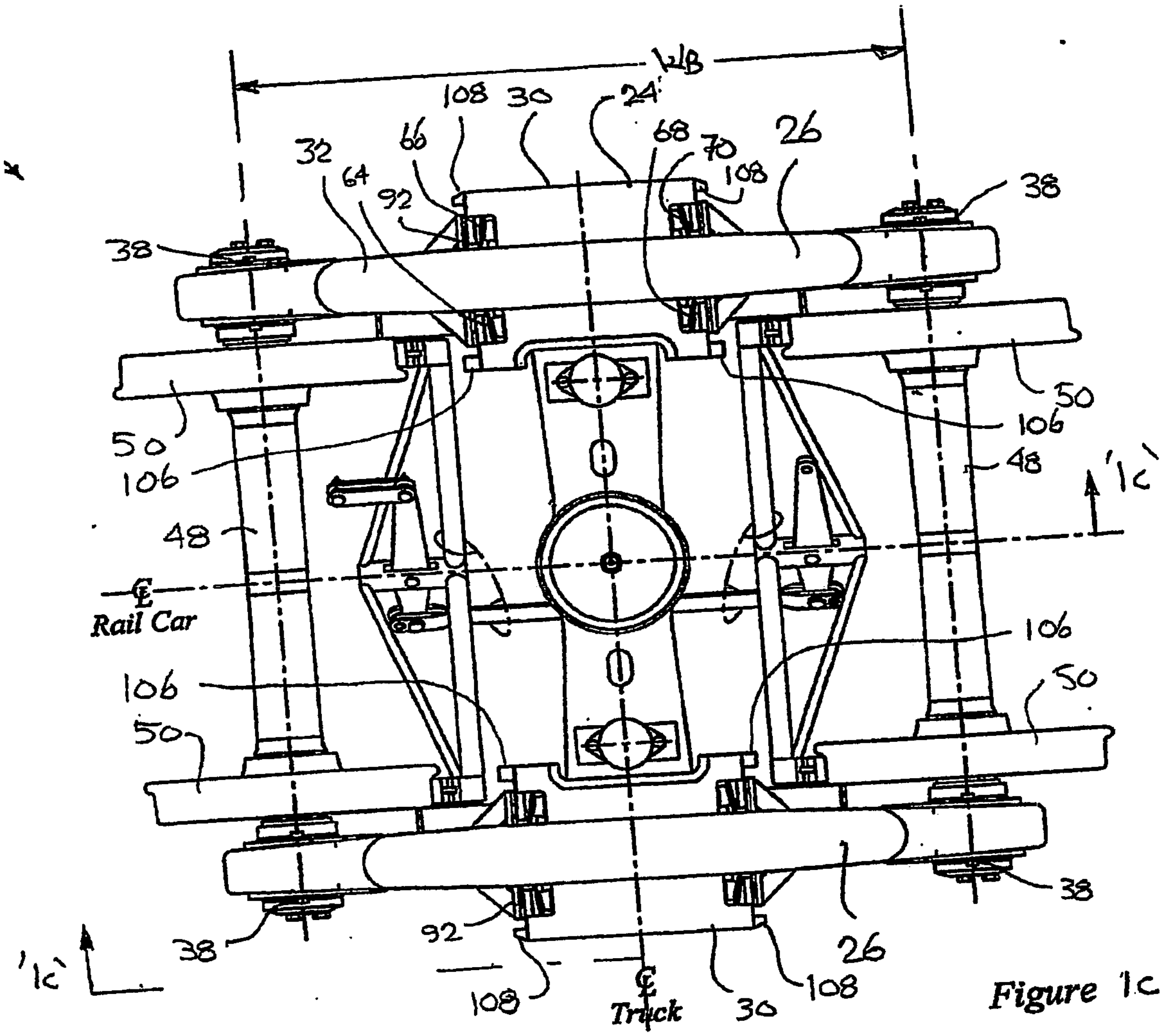

Figure 1c.

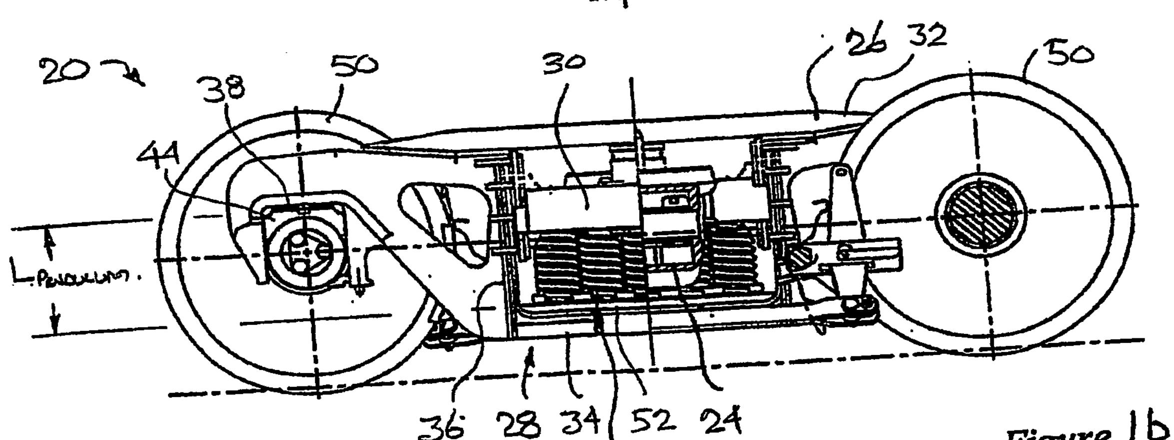

Figure 1b

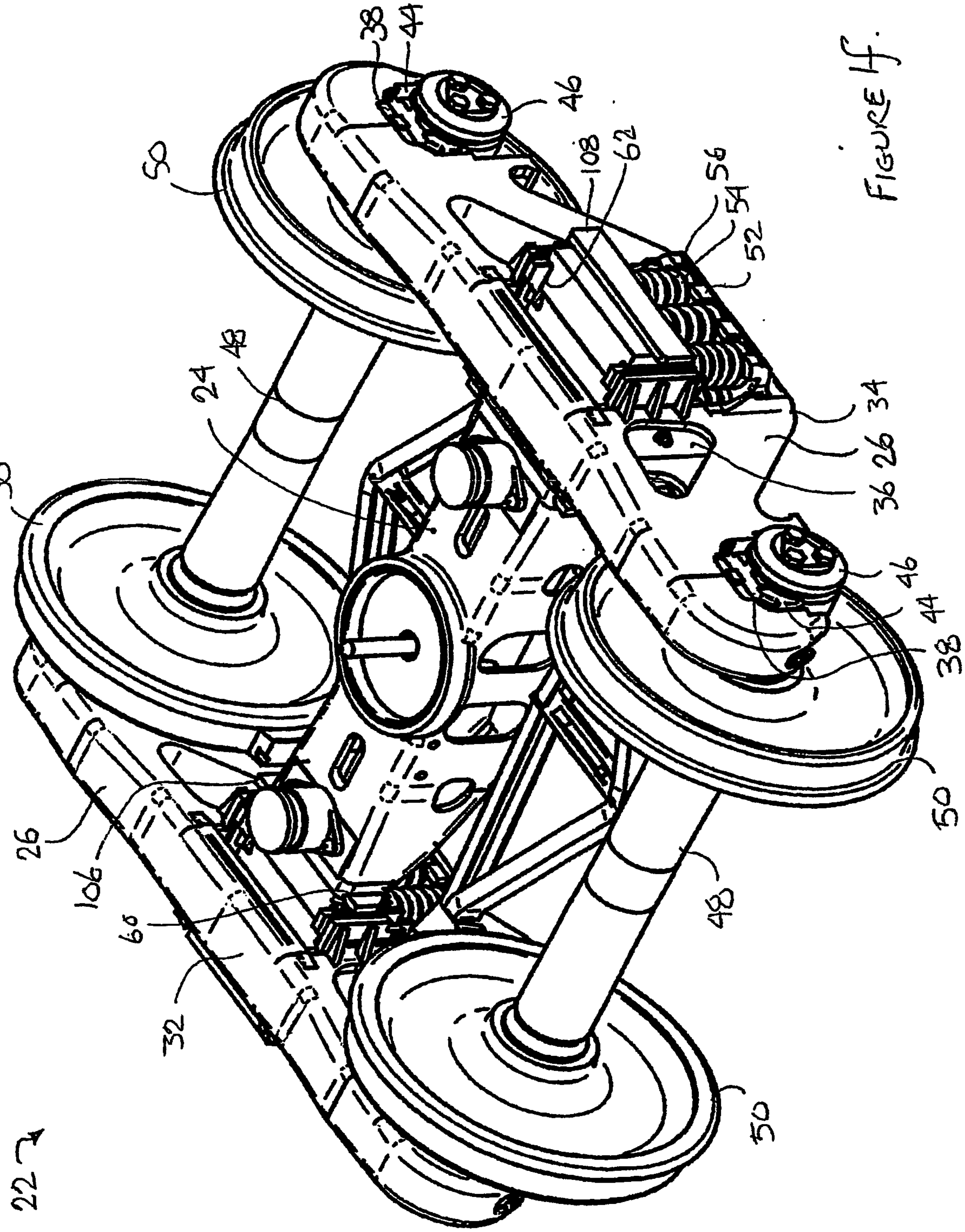
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FIGURE 4.

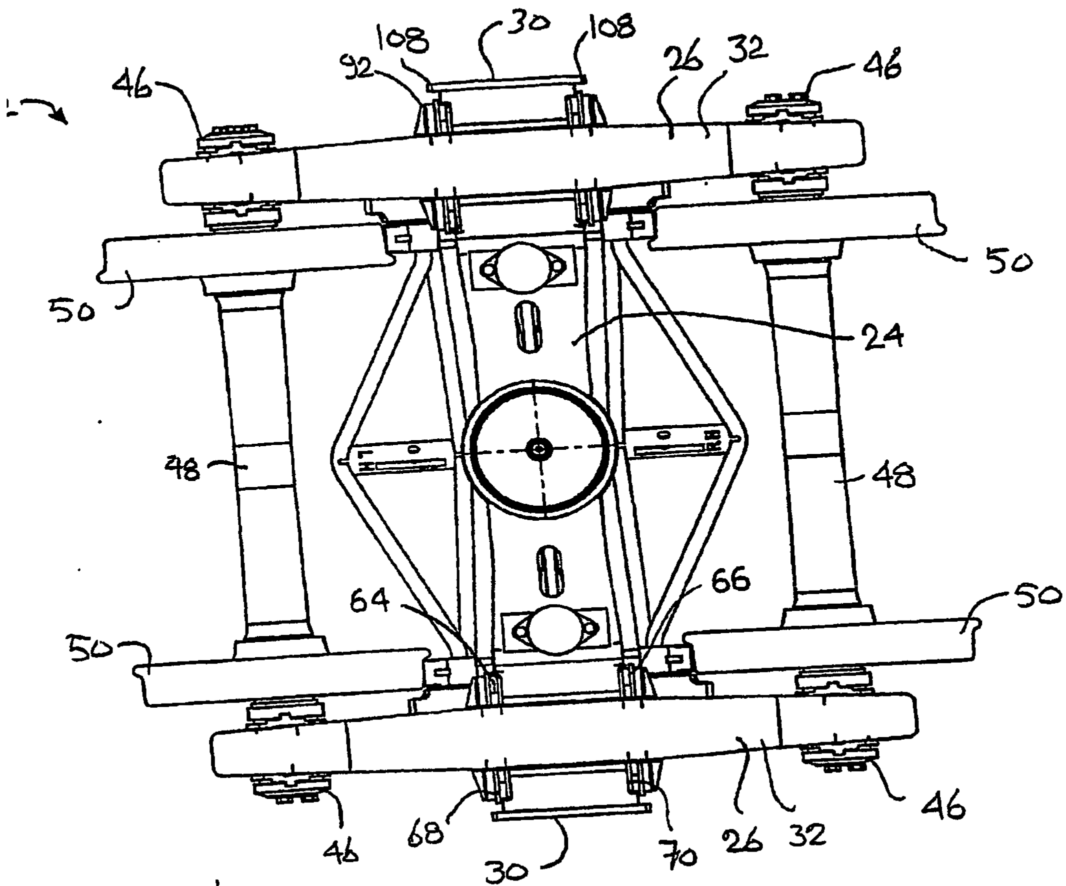

FIGURE 1g

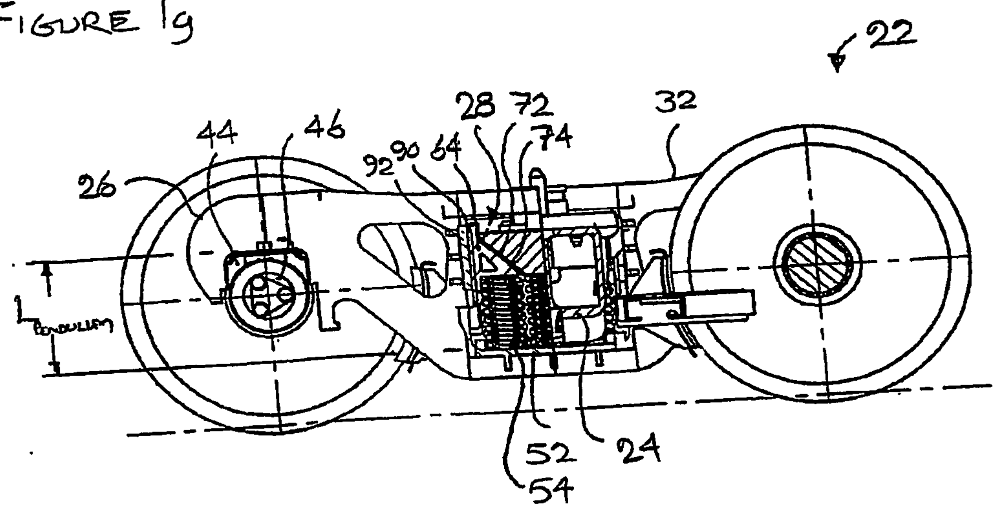

FIGURE 1h.

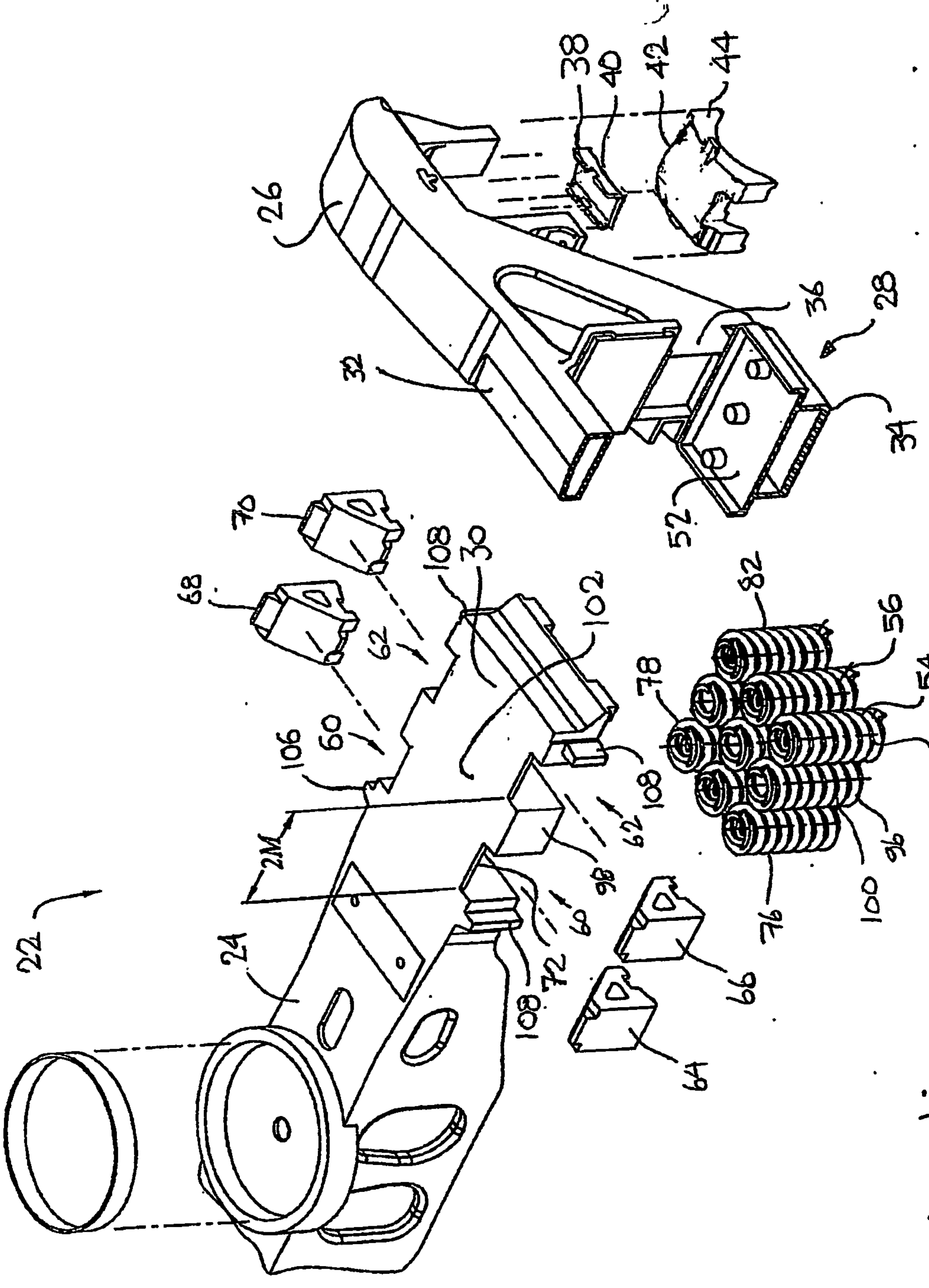
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FIGURE 12

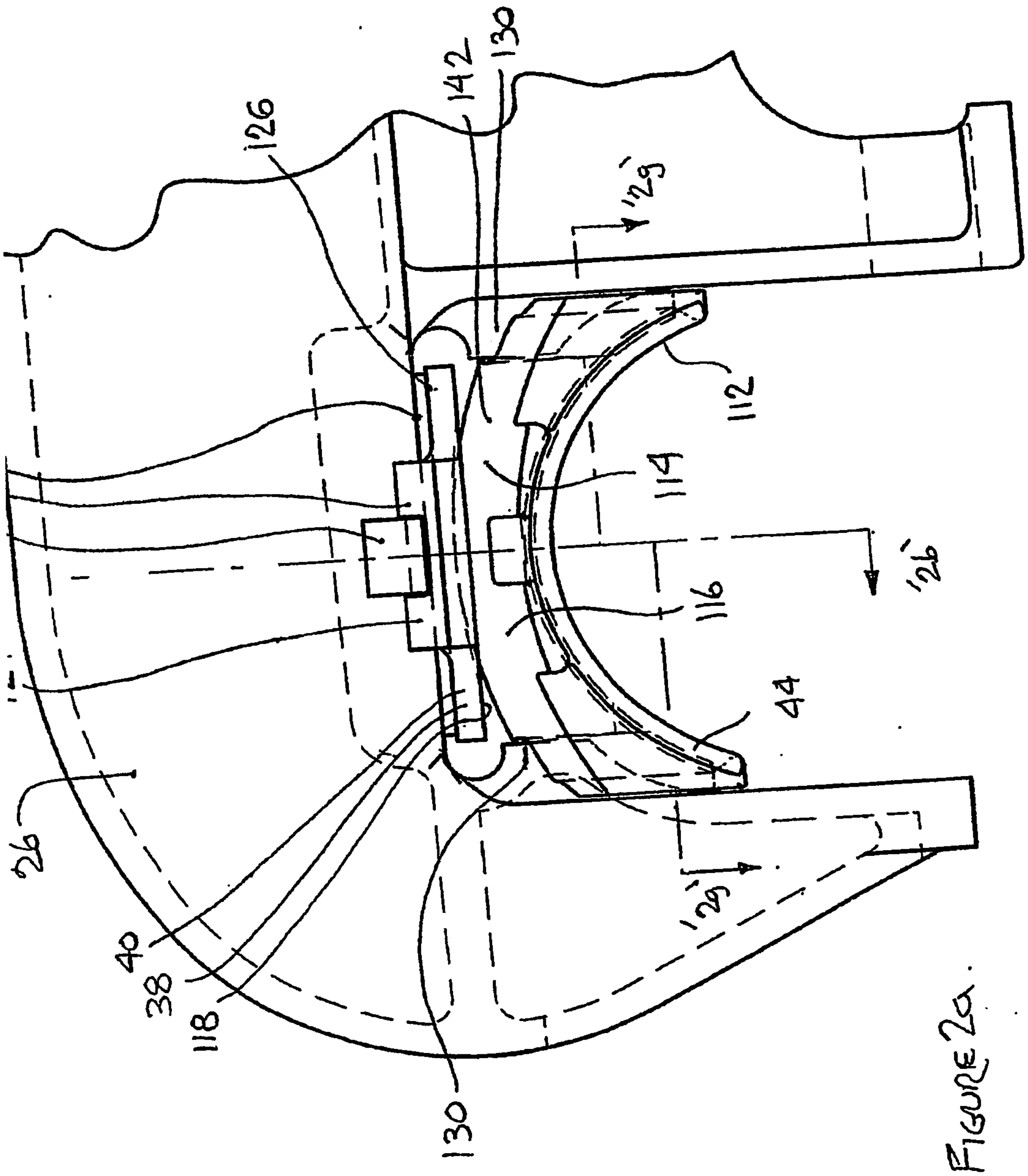


FIGURE 2a

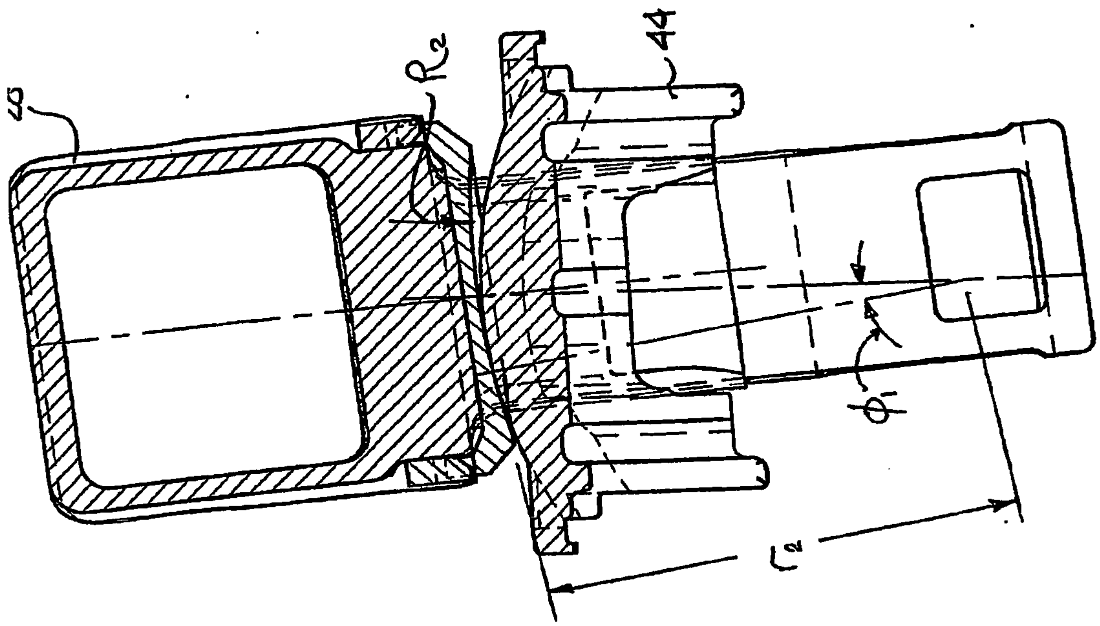


FIGURE 2c.

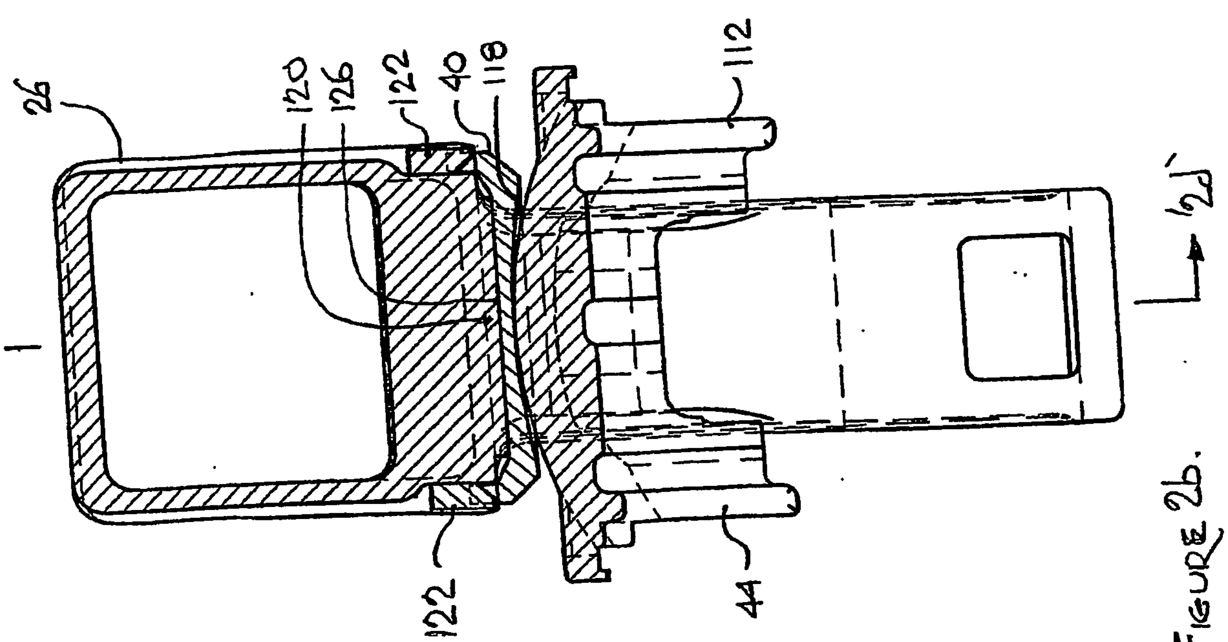


FIGURE 2b.

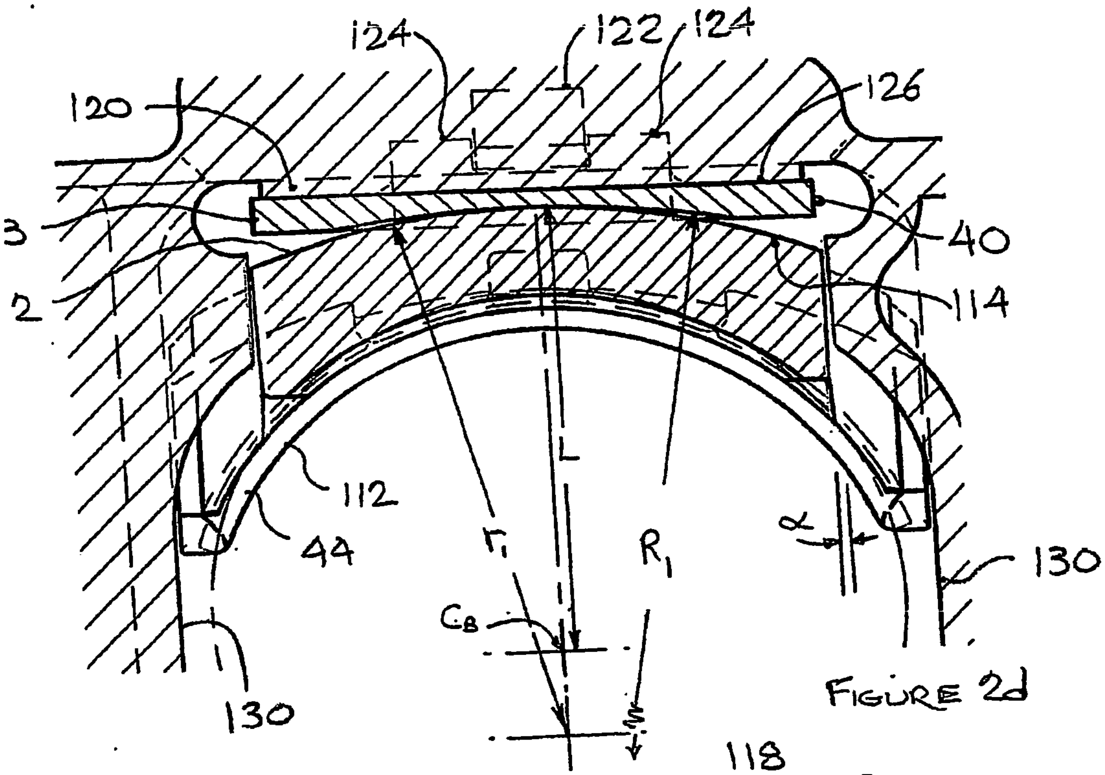


FIGURE 2d

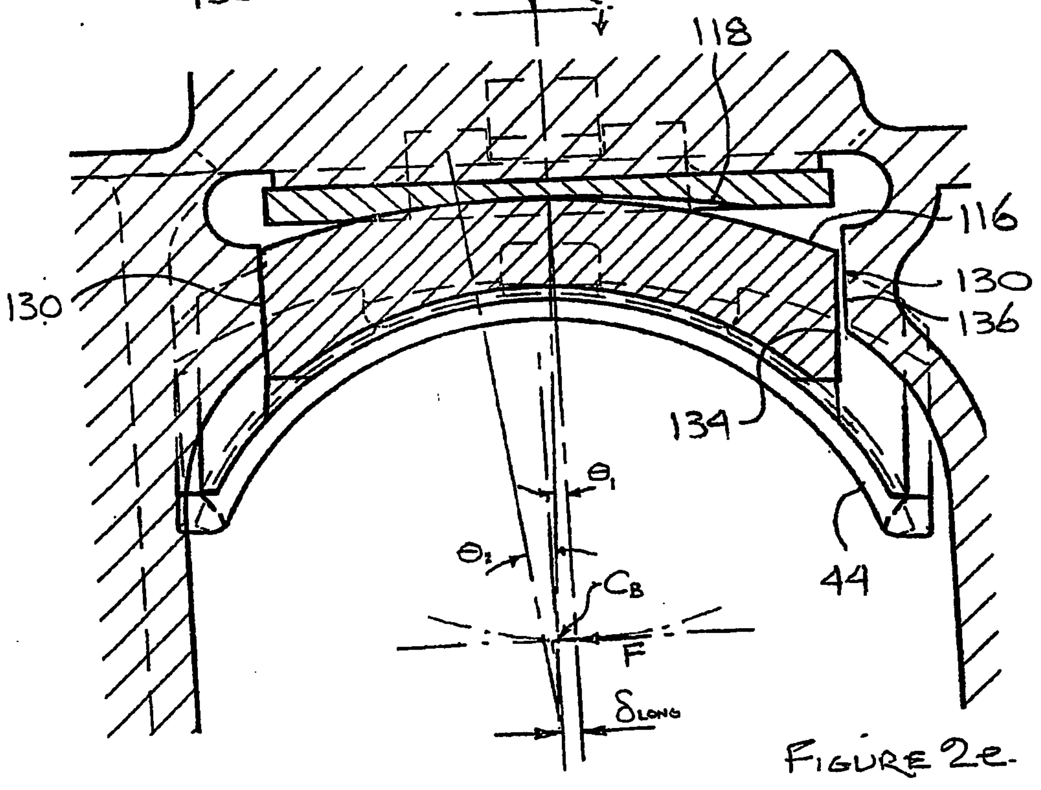
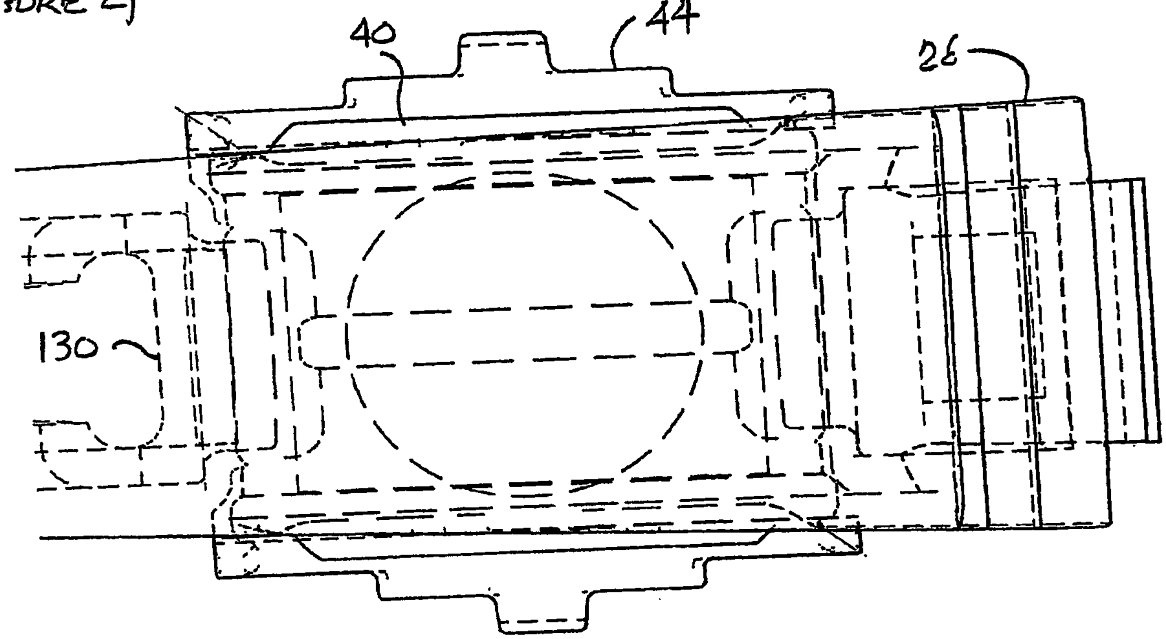


FIGURE 2e

URE 2f



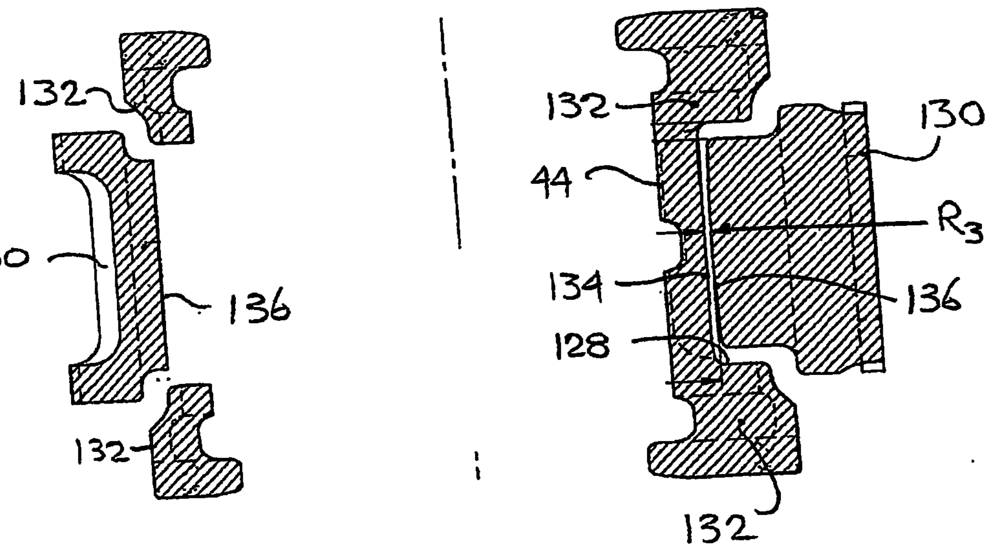
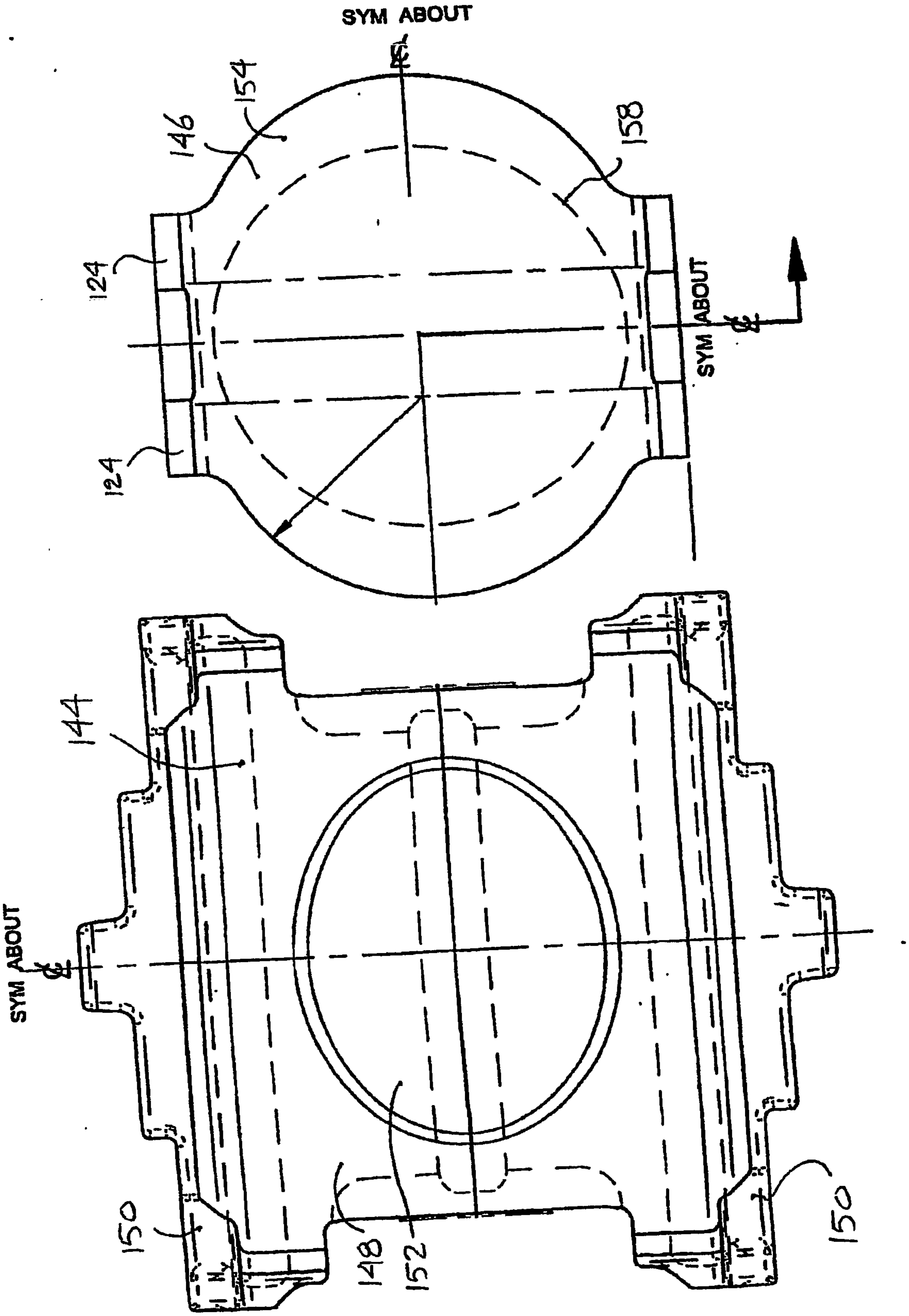


FIGURE 2g.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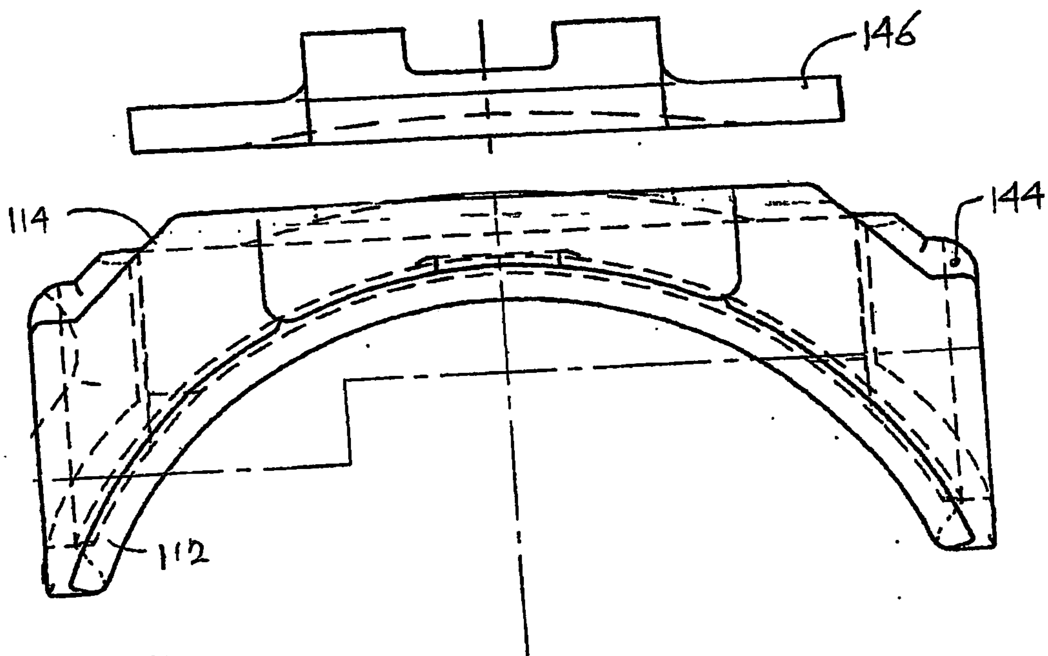
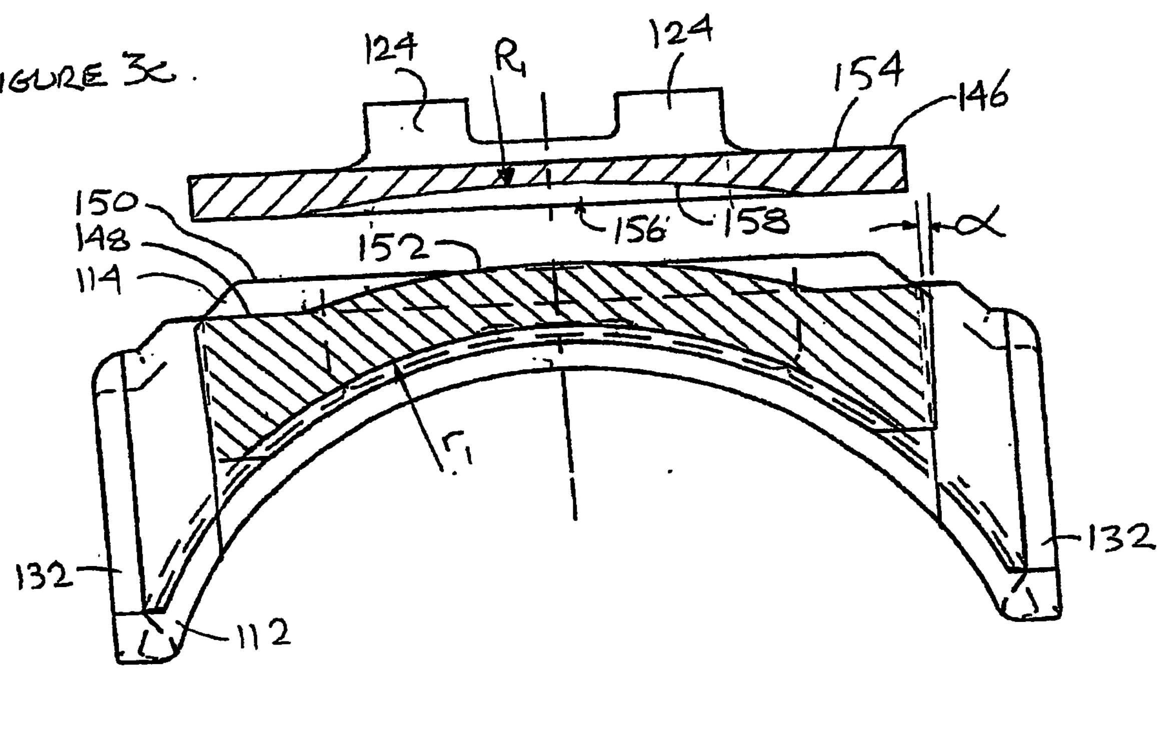
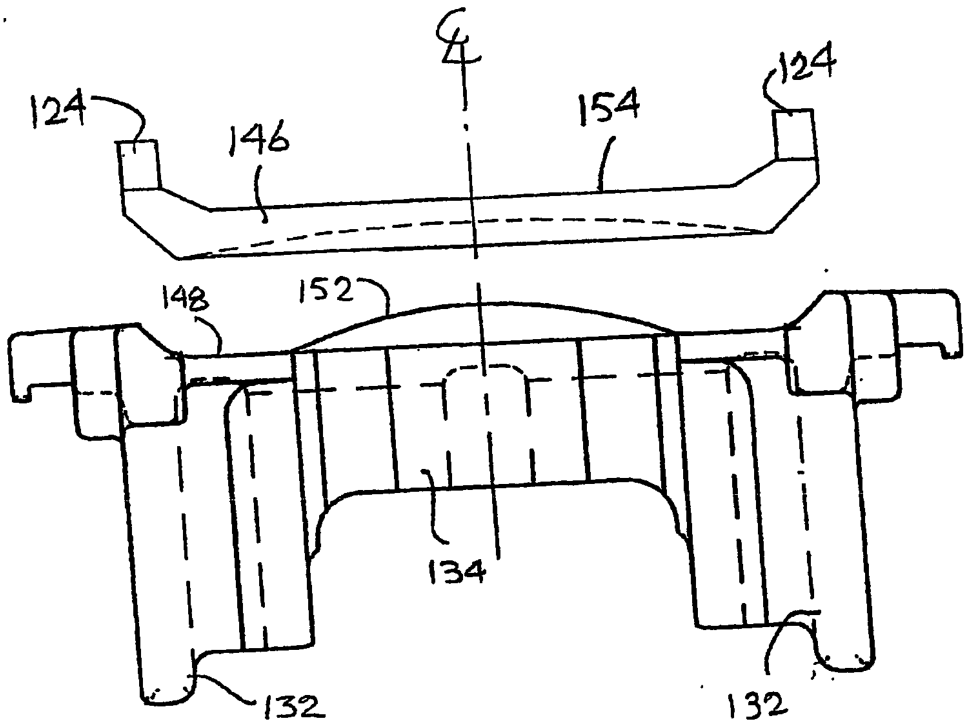


FIGURE 3b

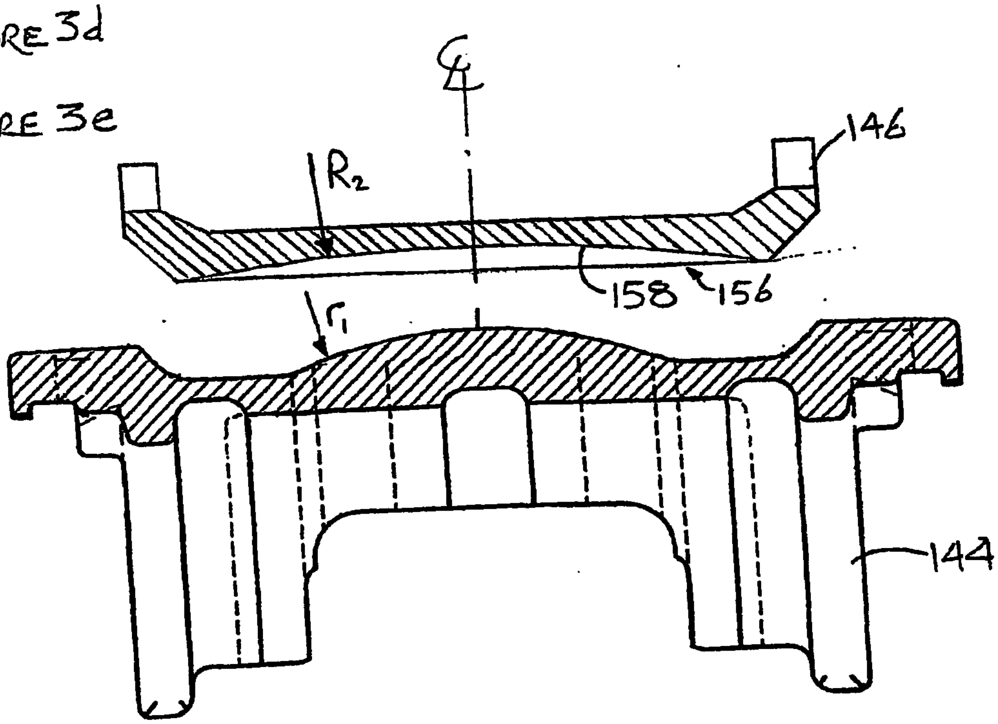
FIGURE 3c.

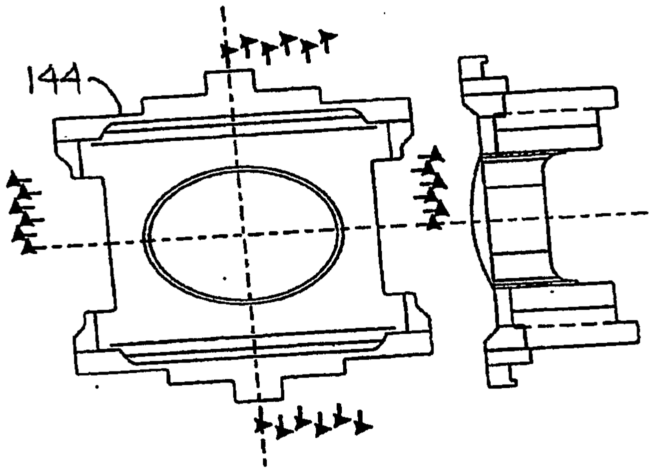


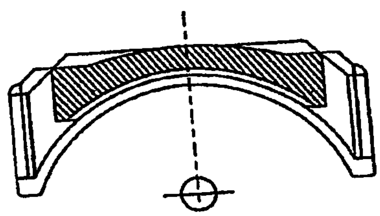


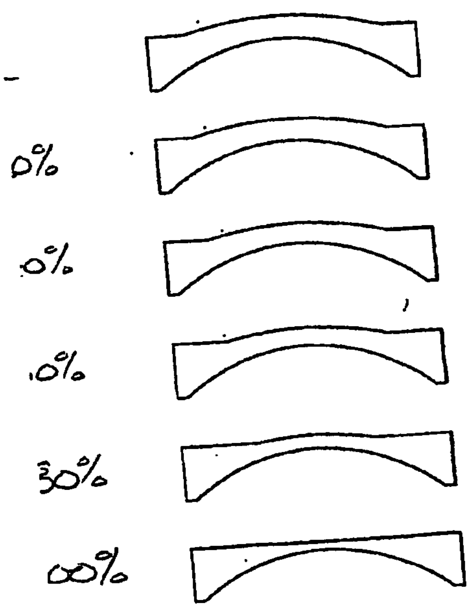
IGURE 3d

IGURE 3e









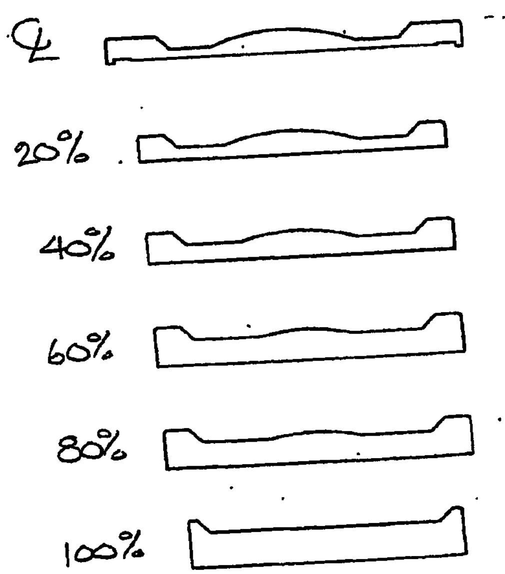


FIGURE 3f

FIGURE 3g

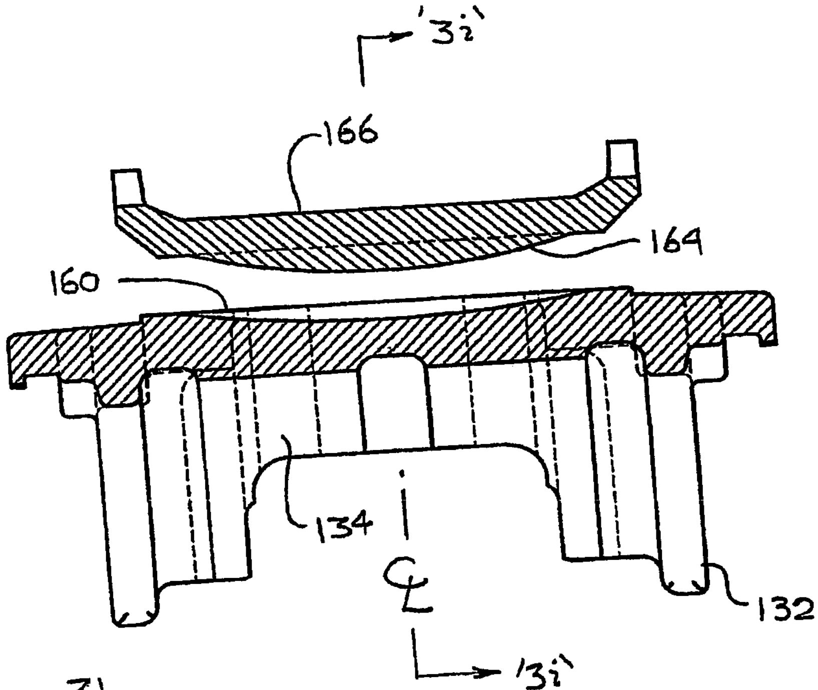


FIGURE 3h

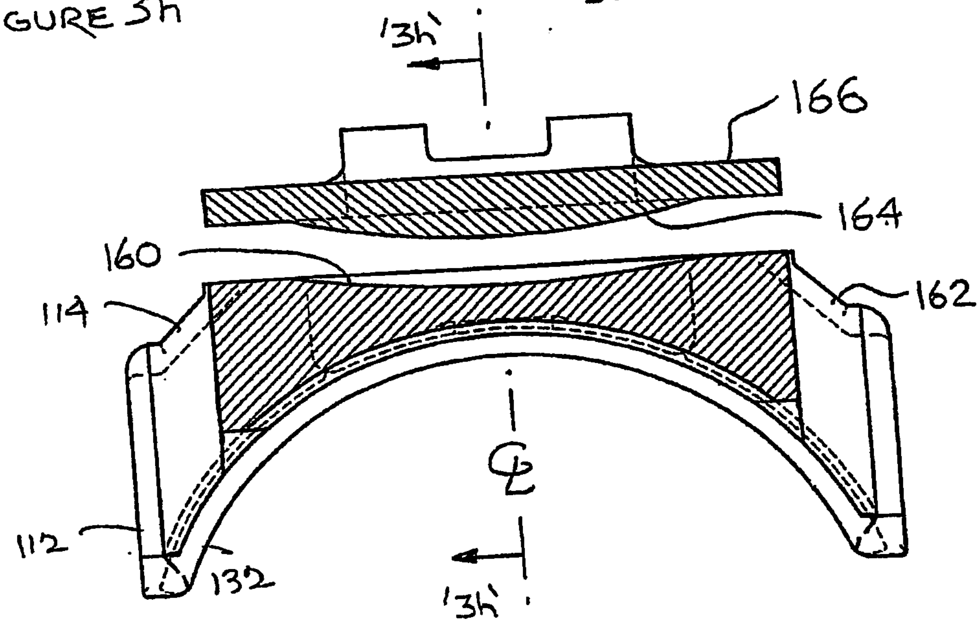


FIGURE 3i

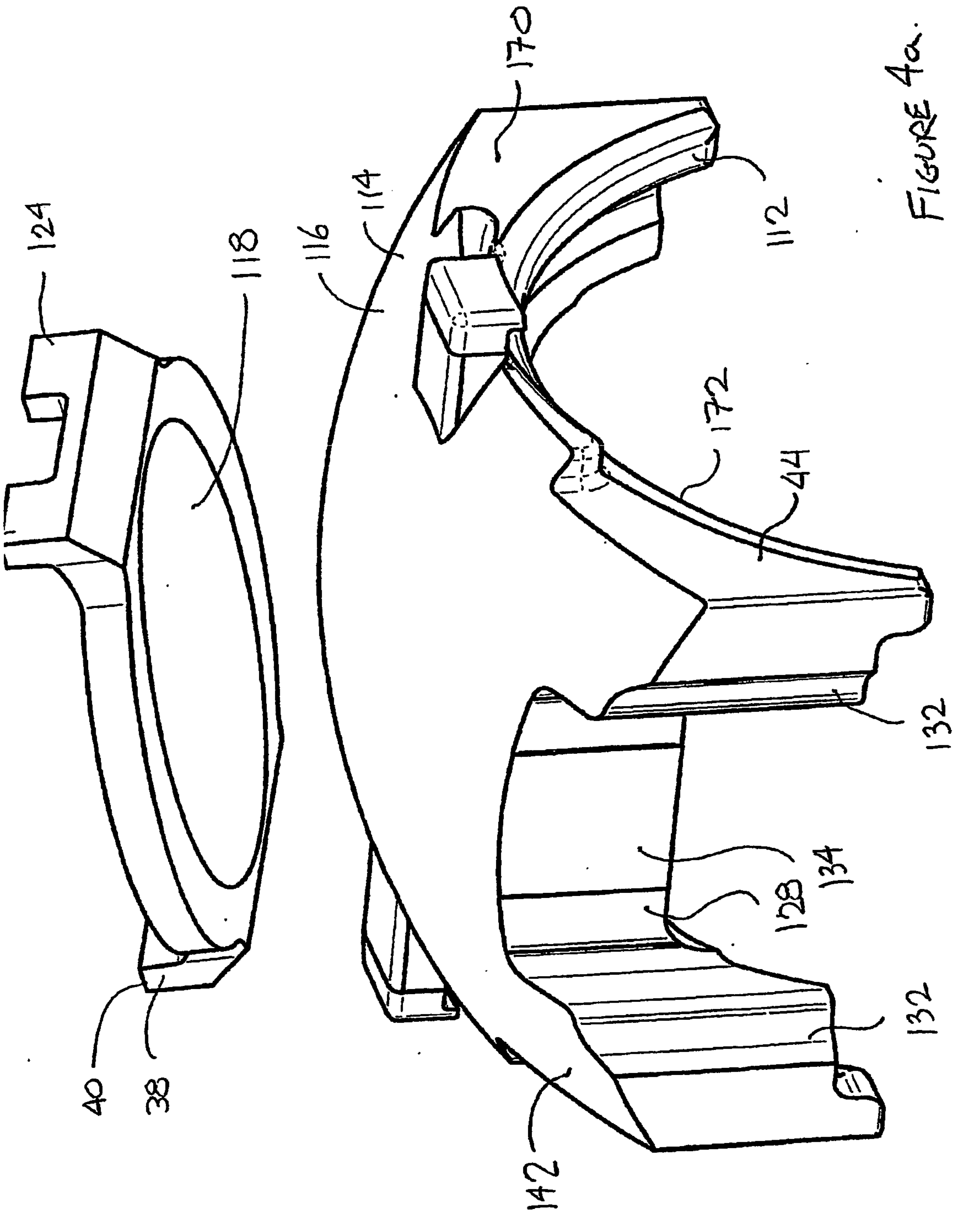


FIGURE 4a

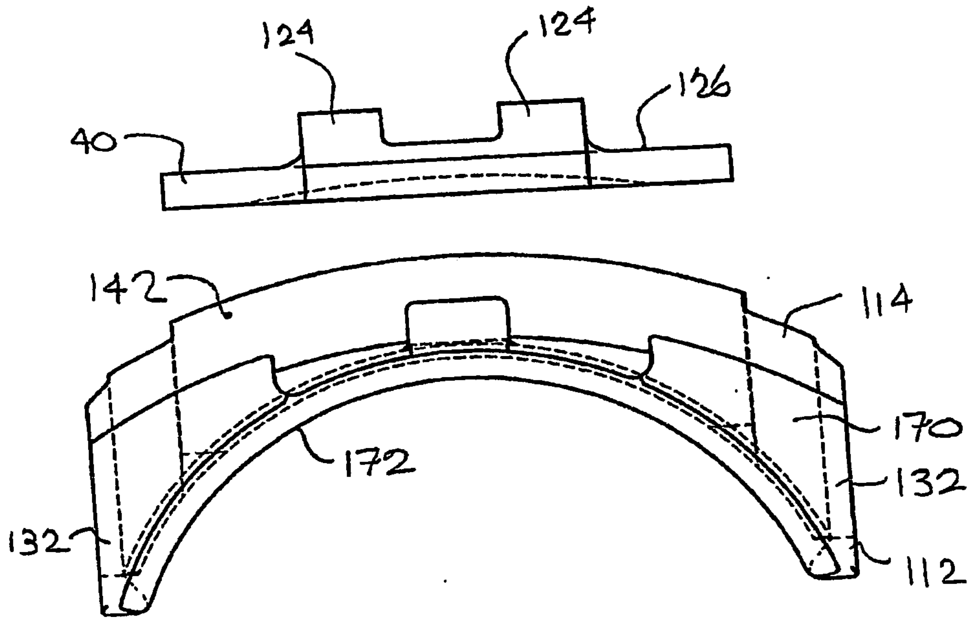


FIGURE 4b

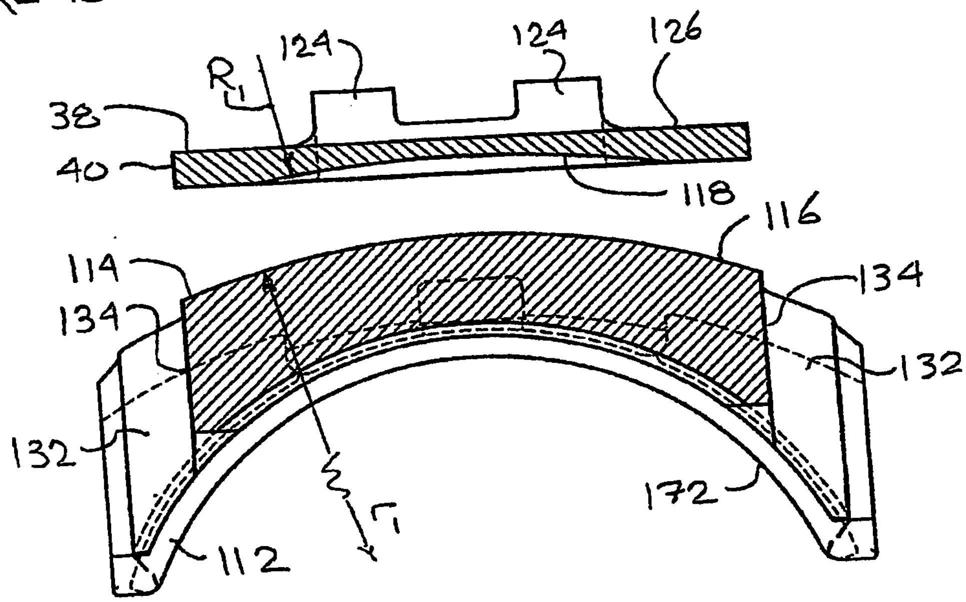


FIGURE 4d.

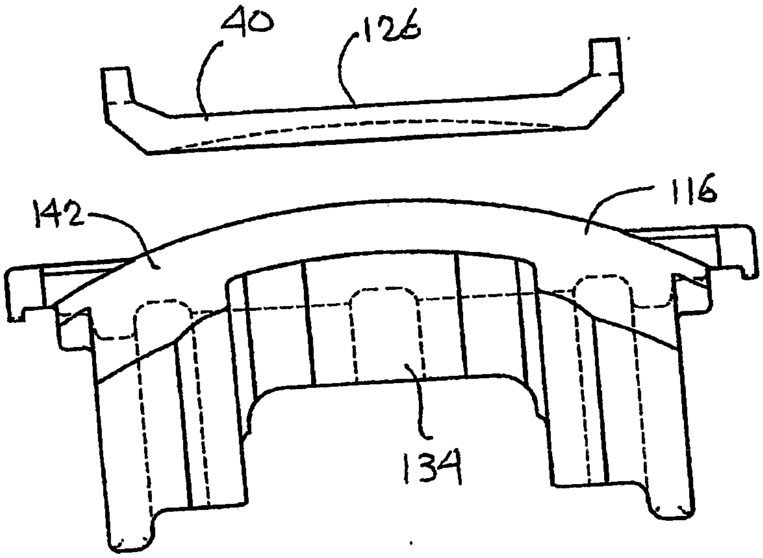


FIGURE 4c.

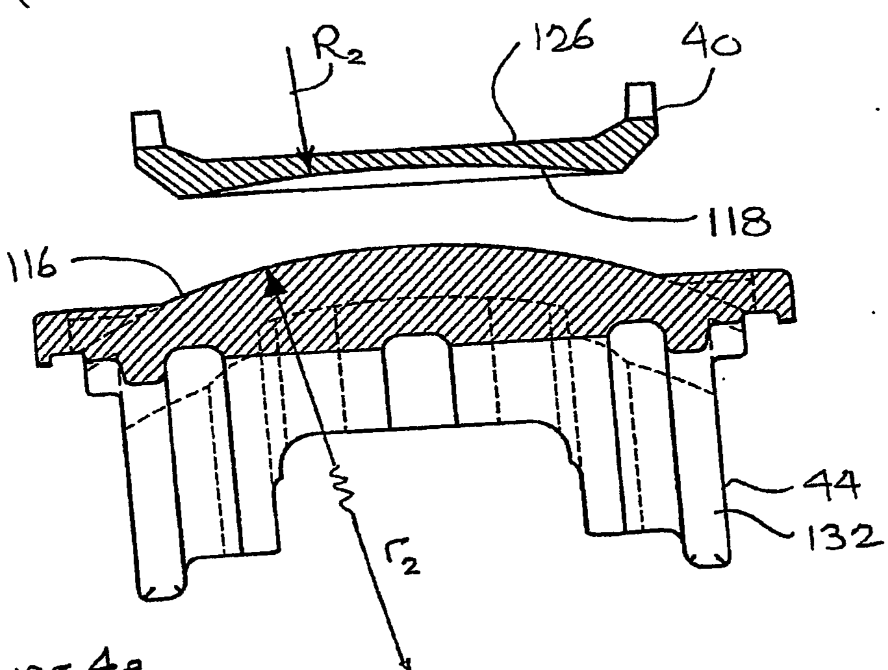


FIGURE 4e

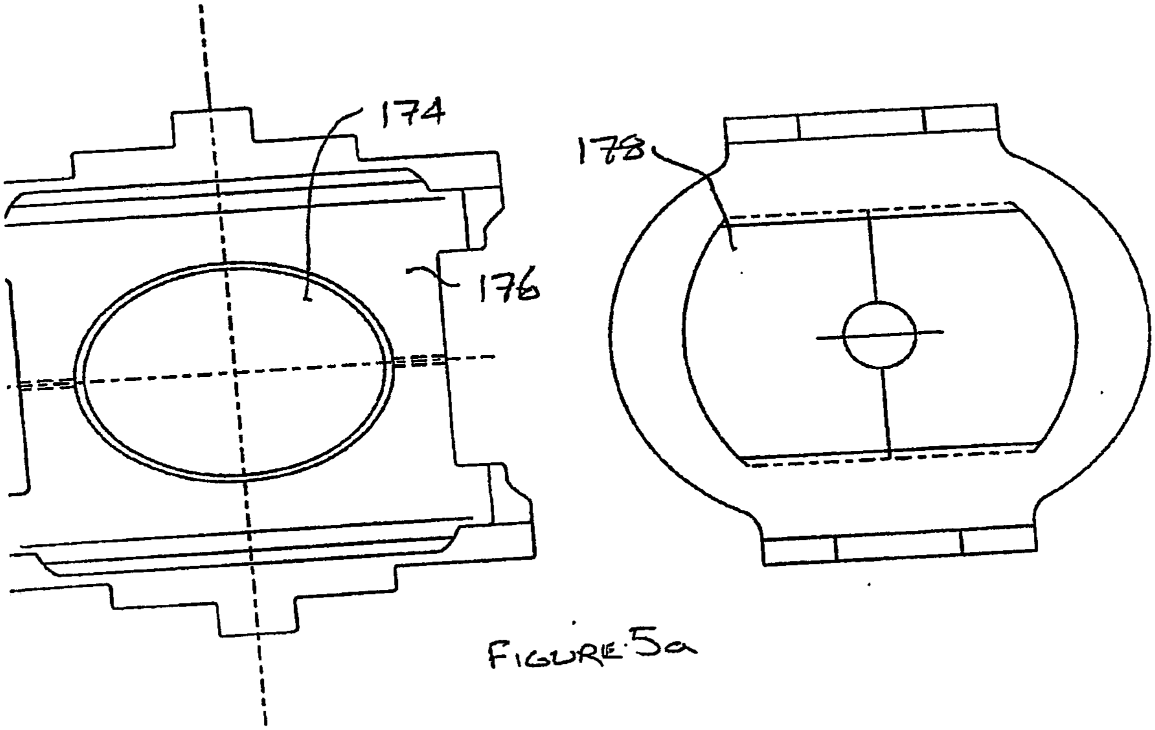


FIGURE 5a

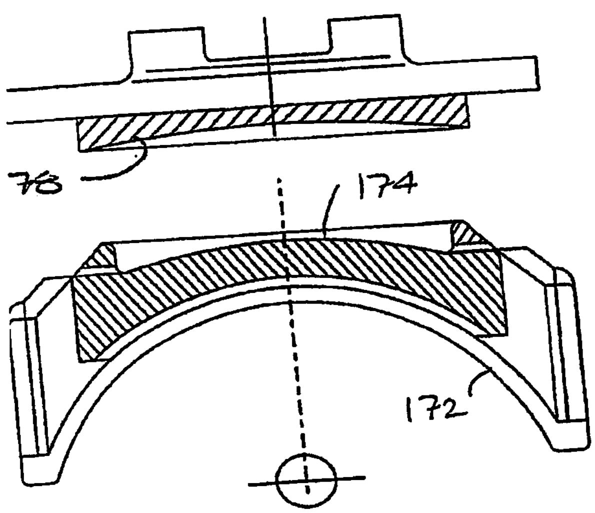


FIGURE 5b

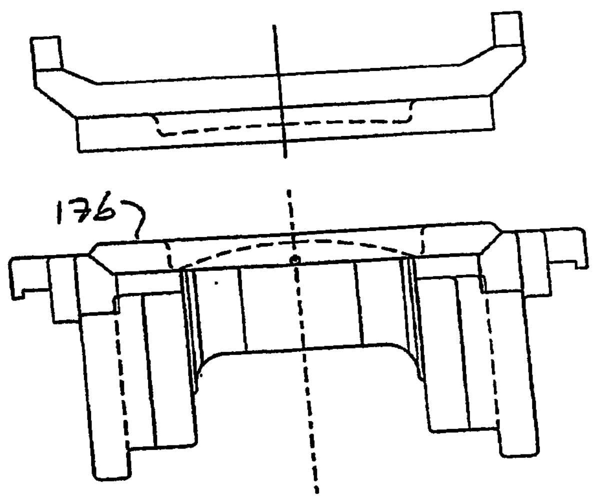


FIGURE 5c

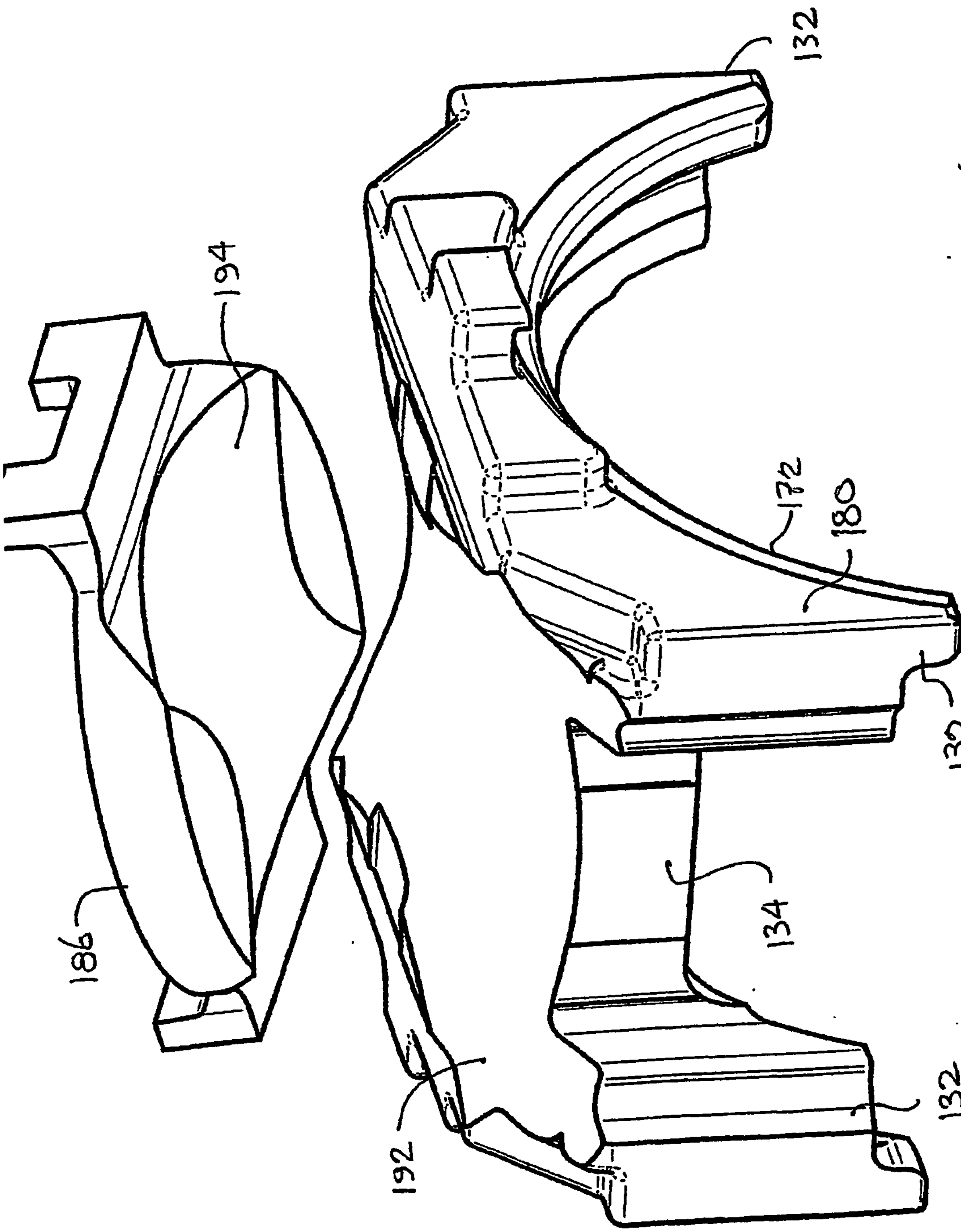


FIGURE 6a.

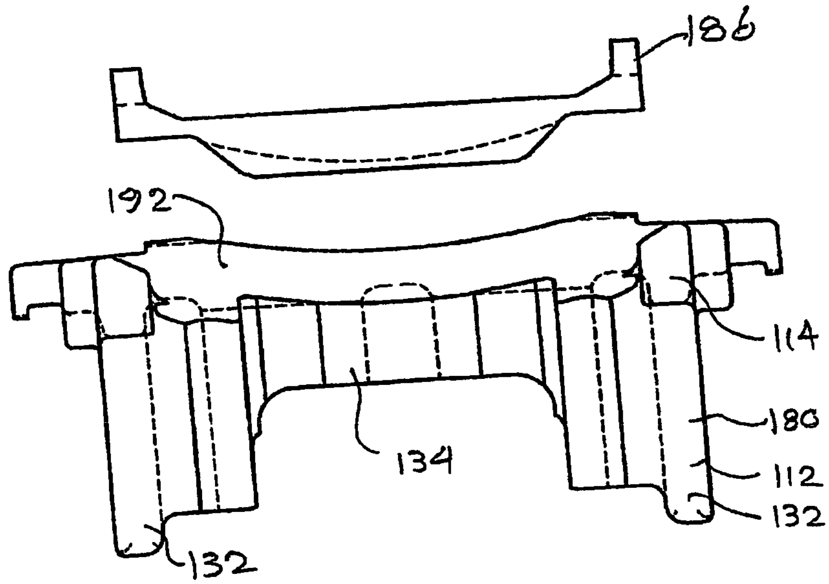


FIGURE 6b.

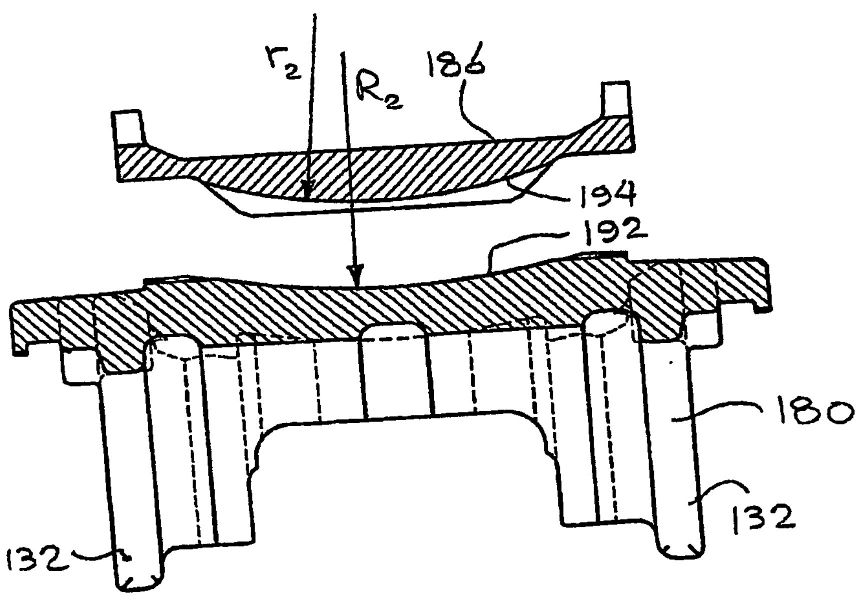


FIGURE 6d.

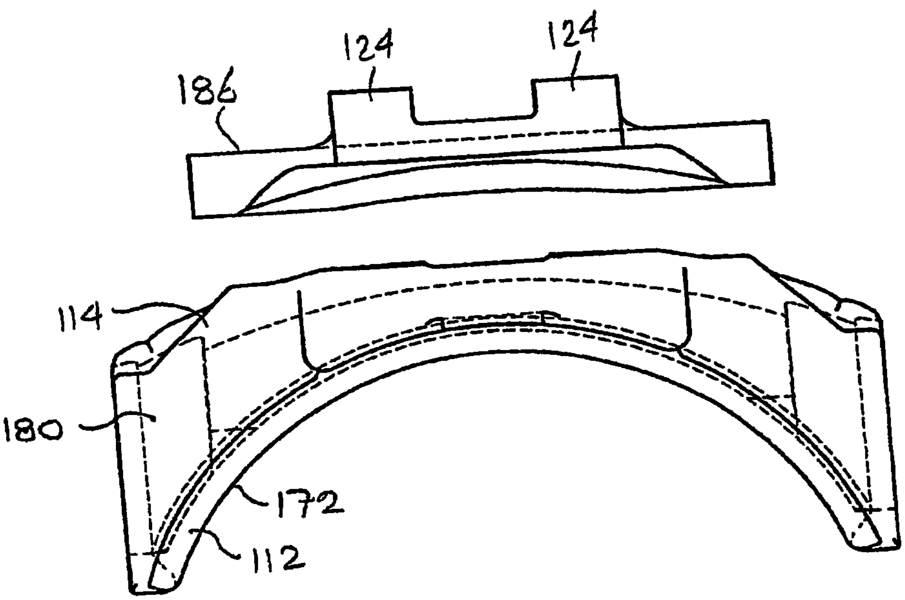


FIGURE 6C.

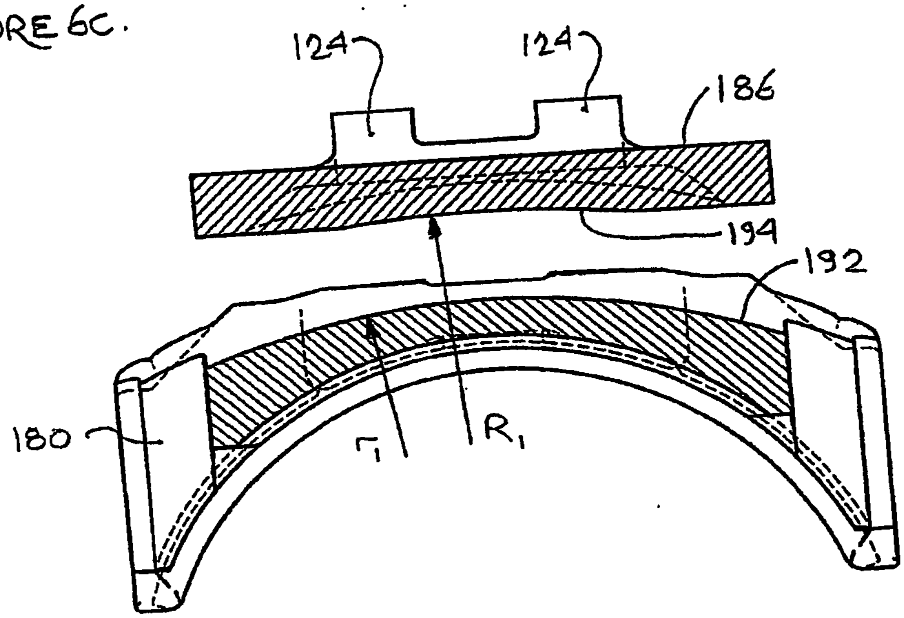


FIGURE 6e

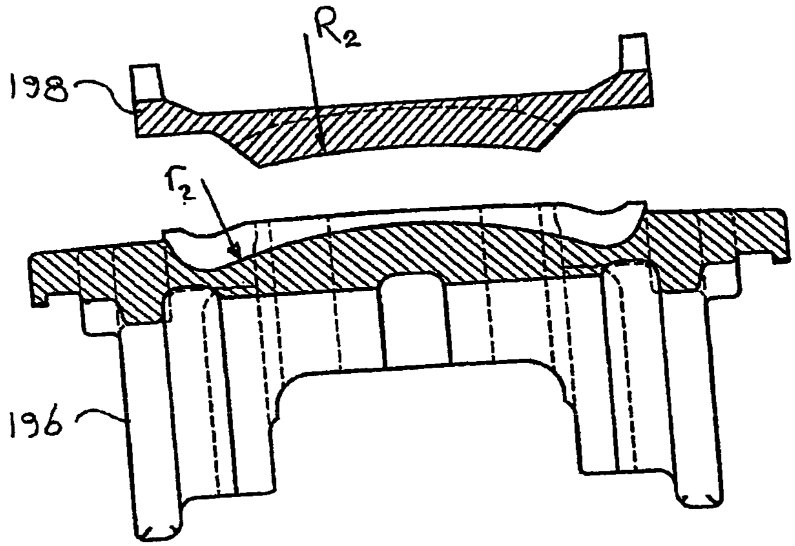


FIGURE 6f

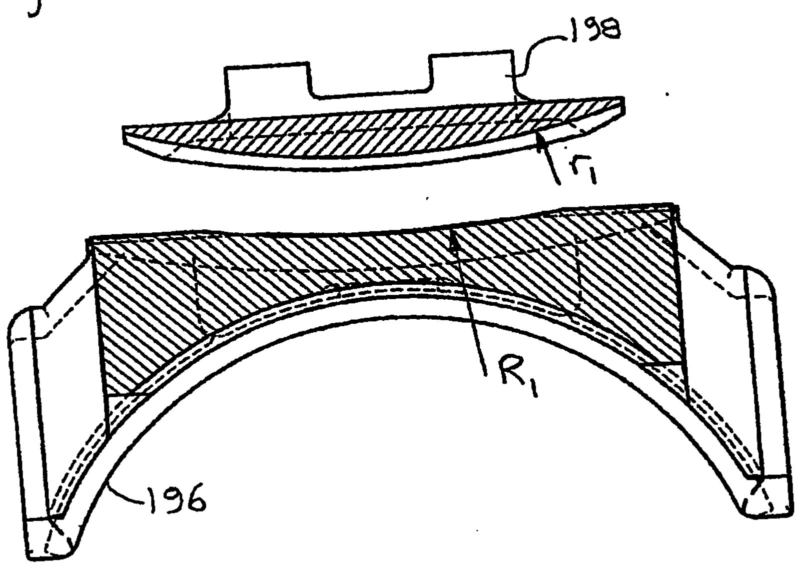


FIGURE 6g

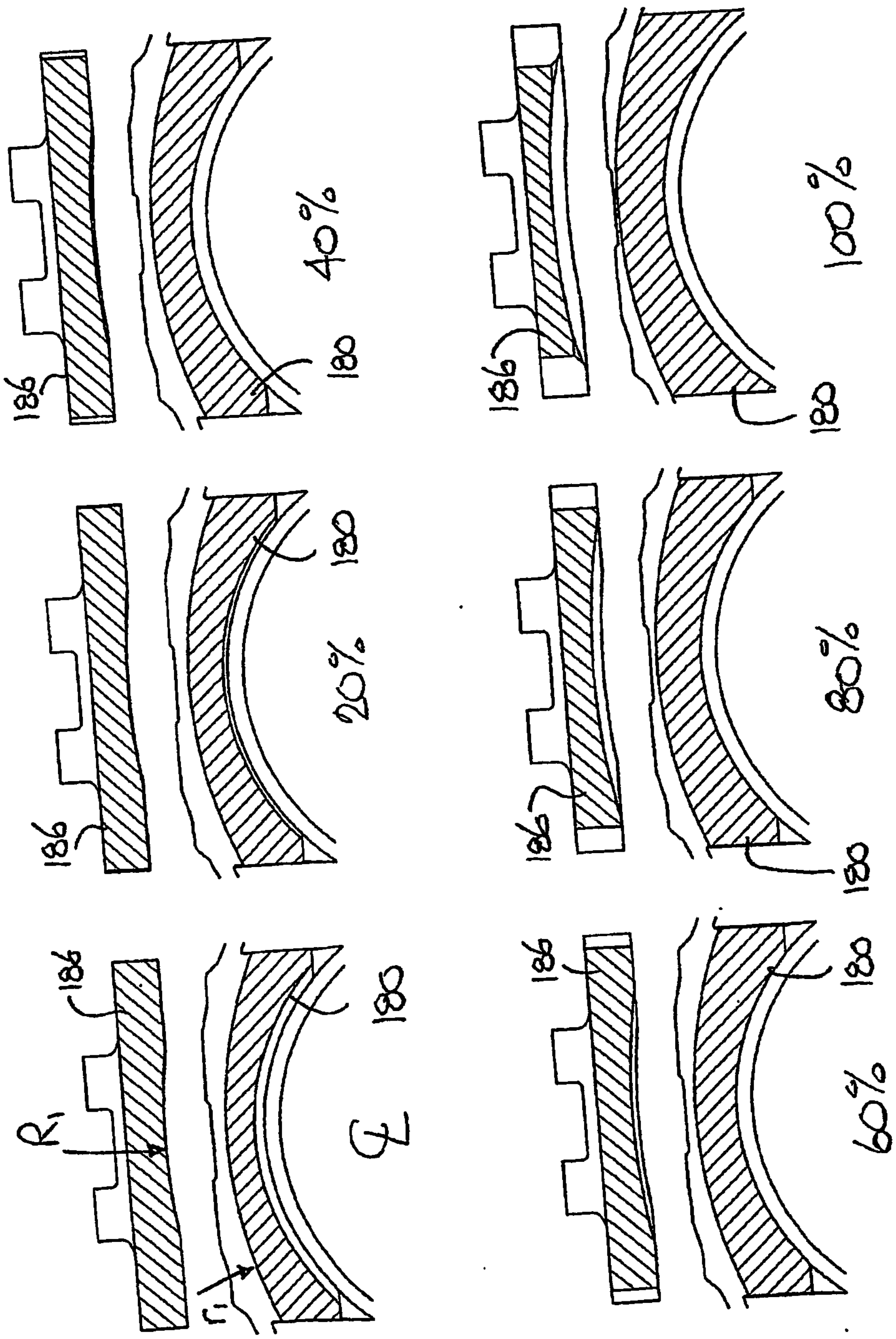


FIGURE 6h

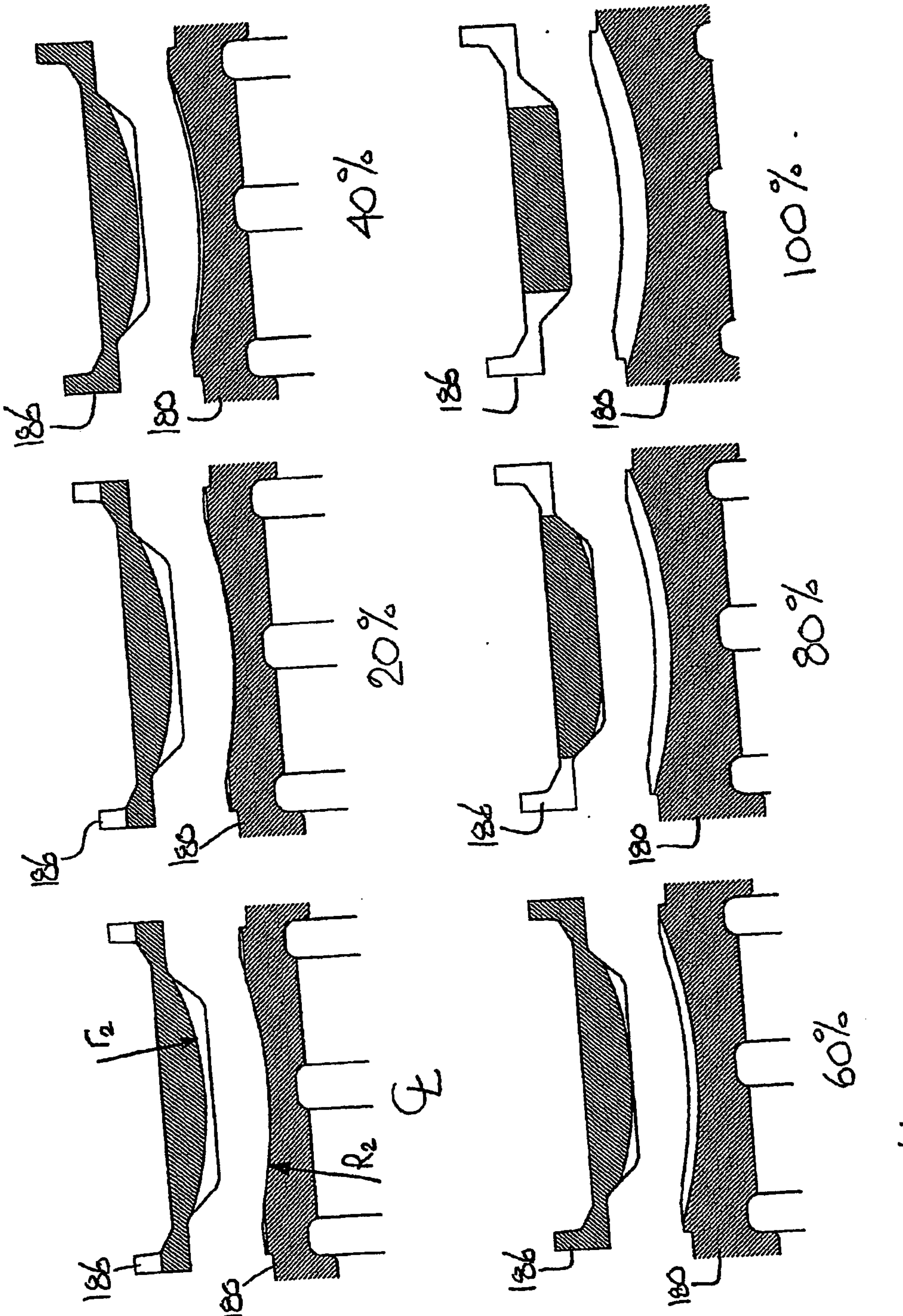


FIGURE 6i

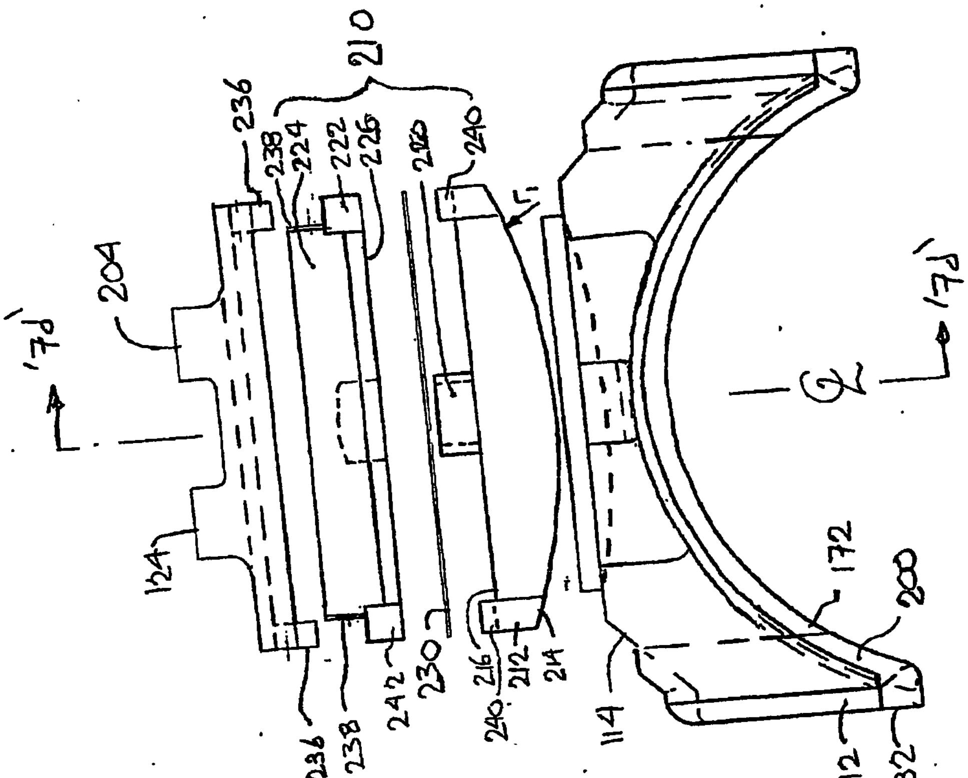


FIGURE 7a

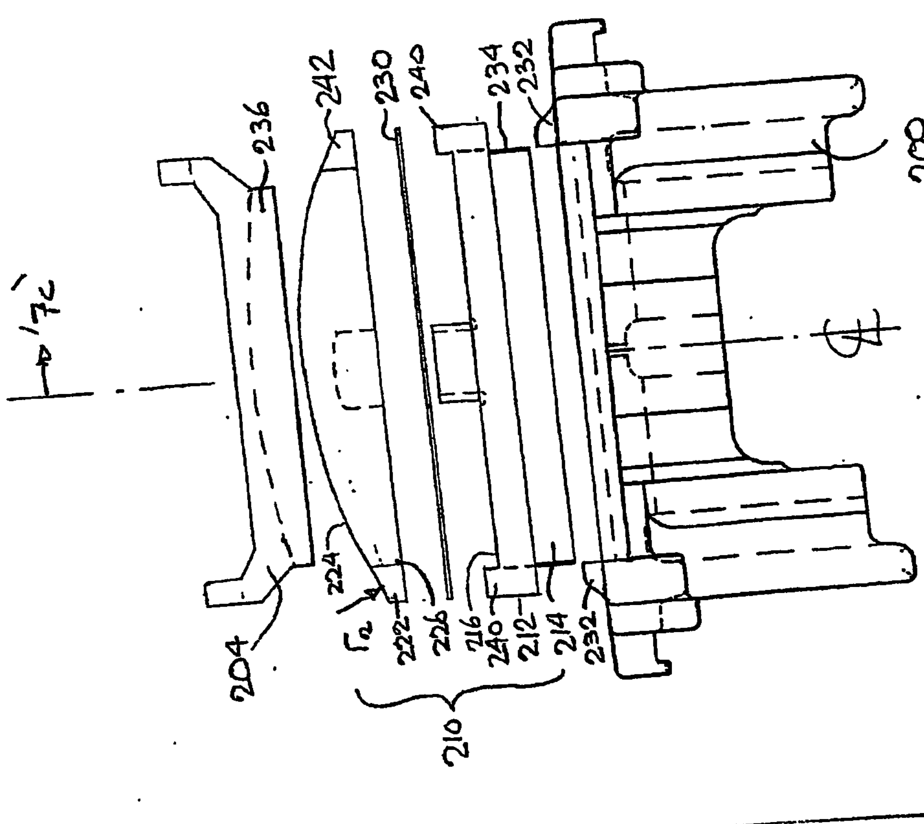


FIGURE 7b

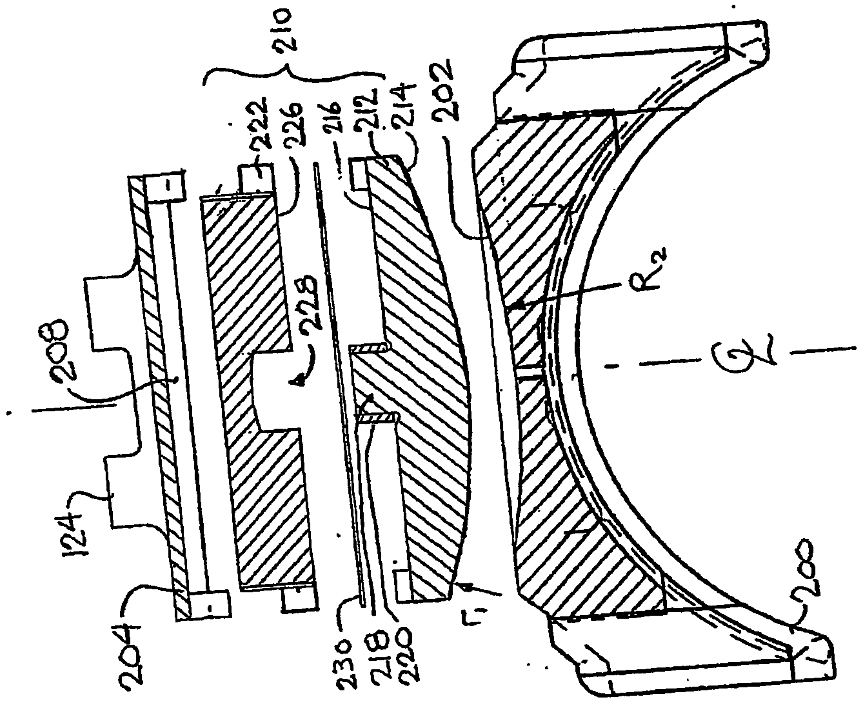


FIGURE 7c.

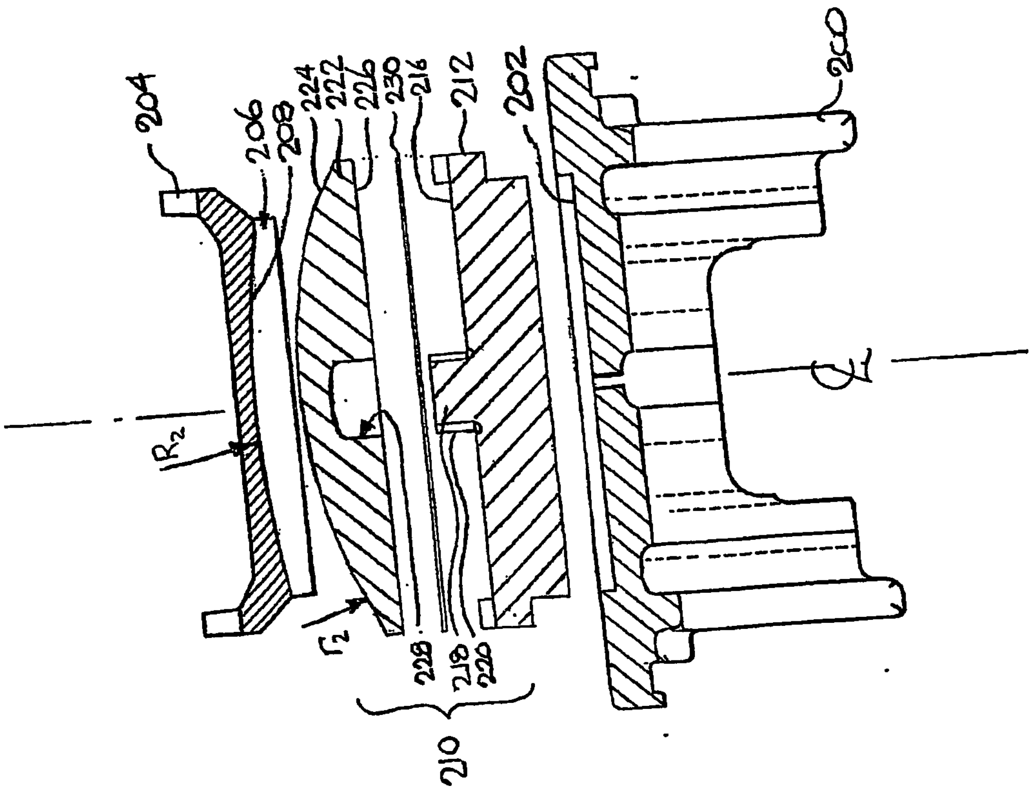


FIGURE 7d

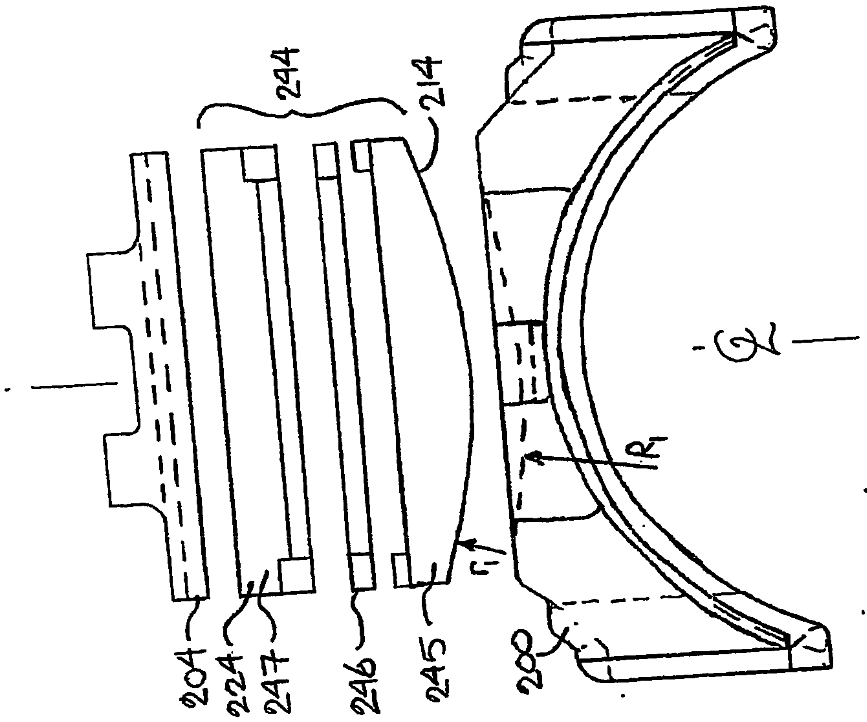


FIGURE 8b

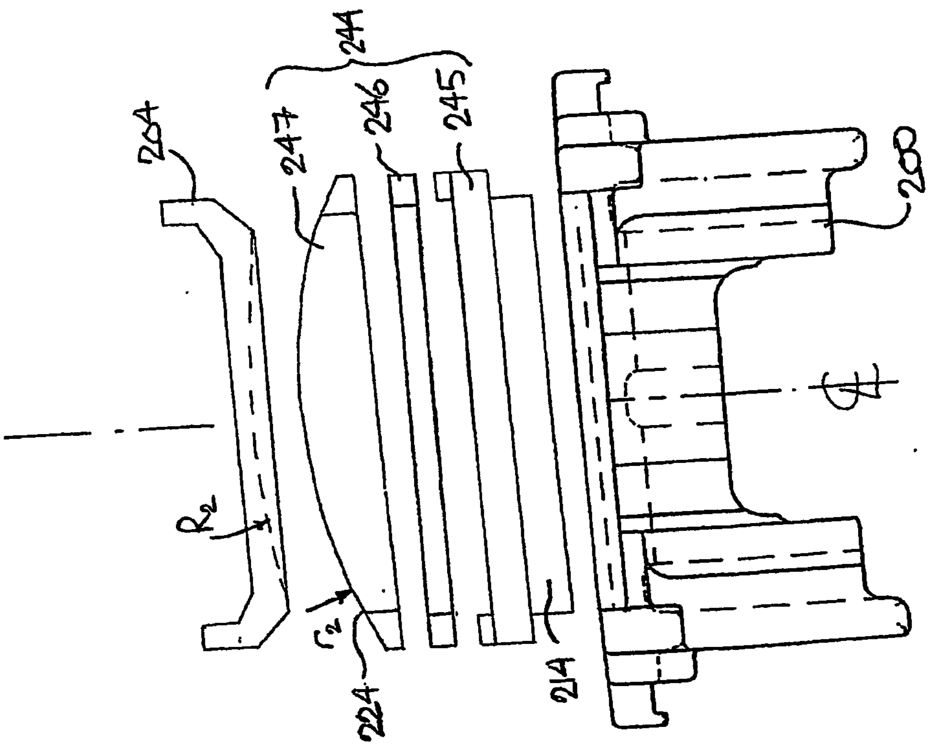


FIGURE 8a

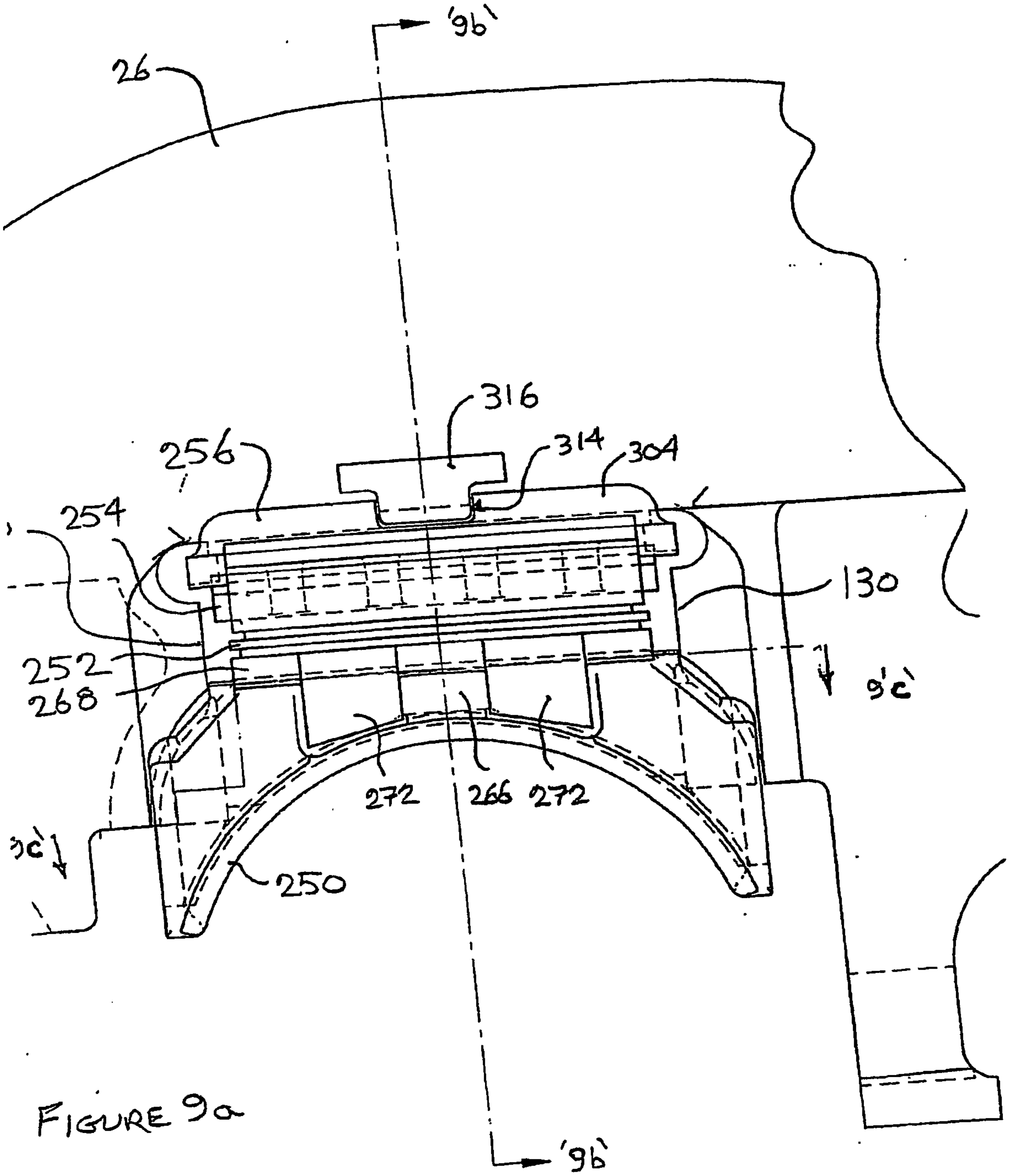


FIGURE 9a

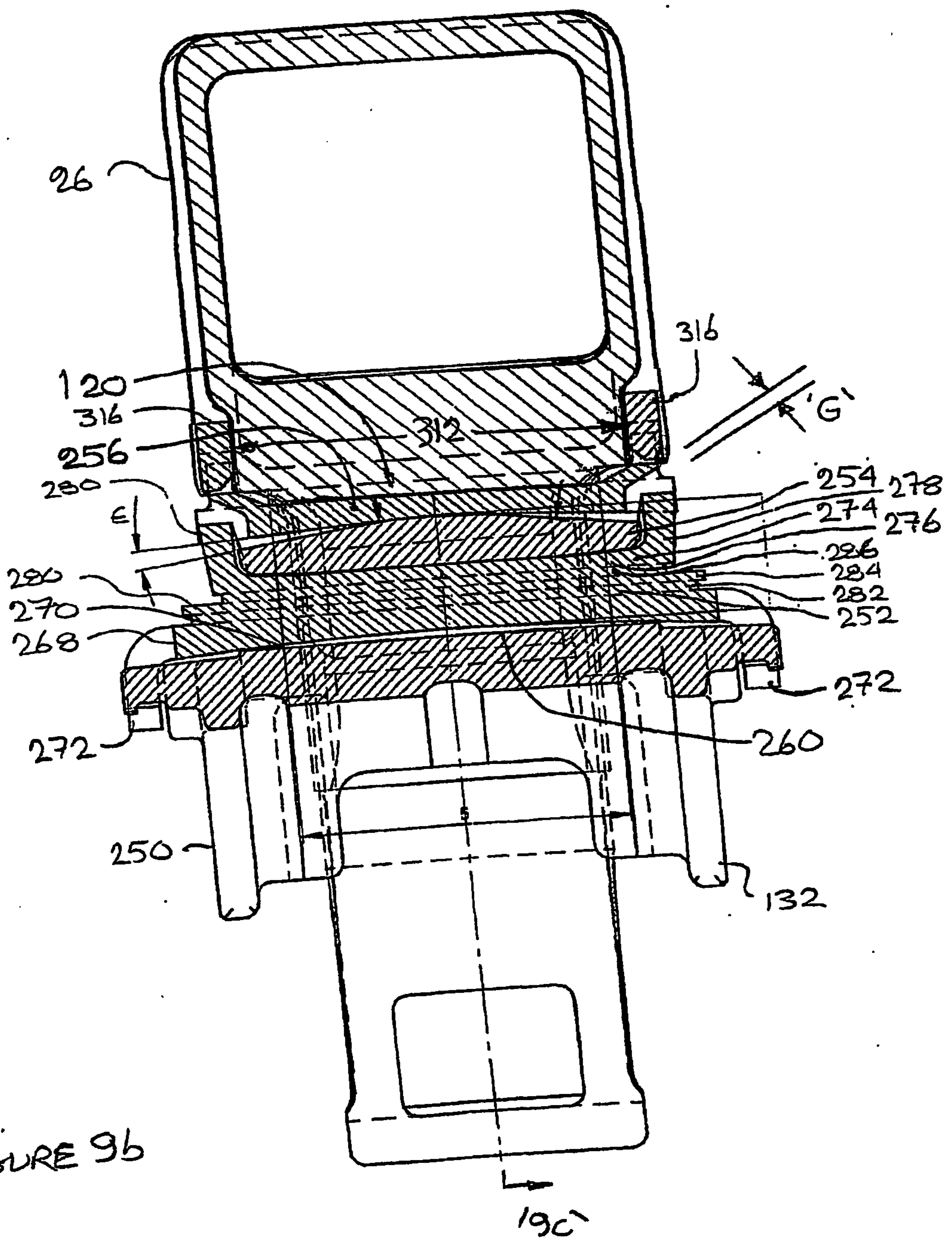

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FIGURE 9b

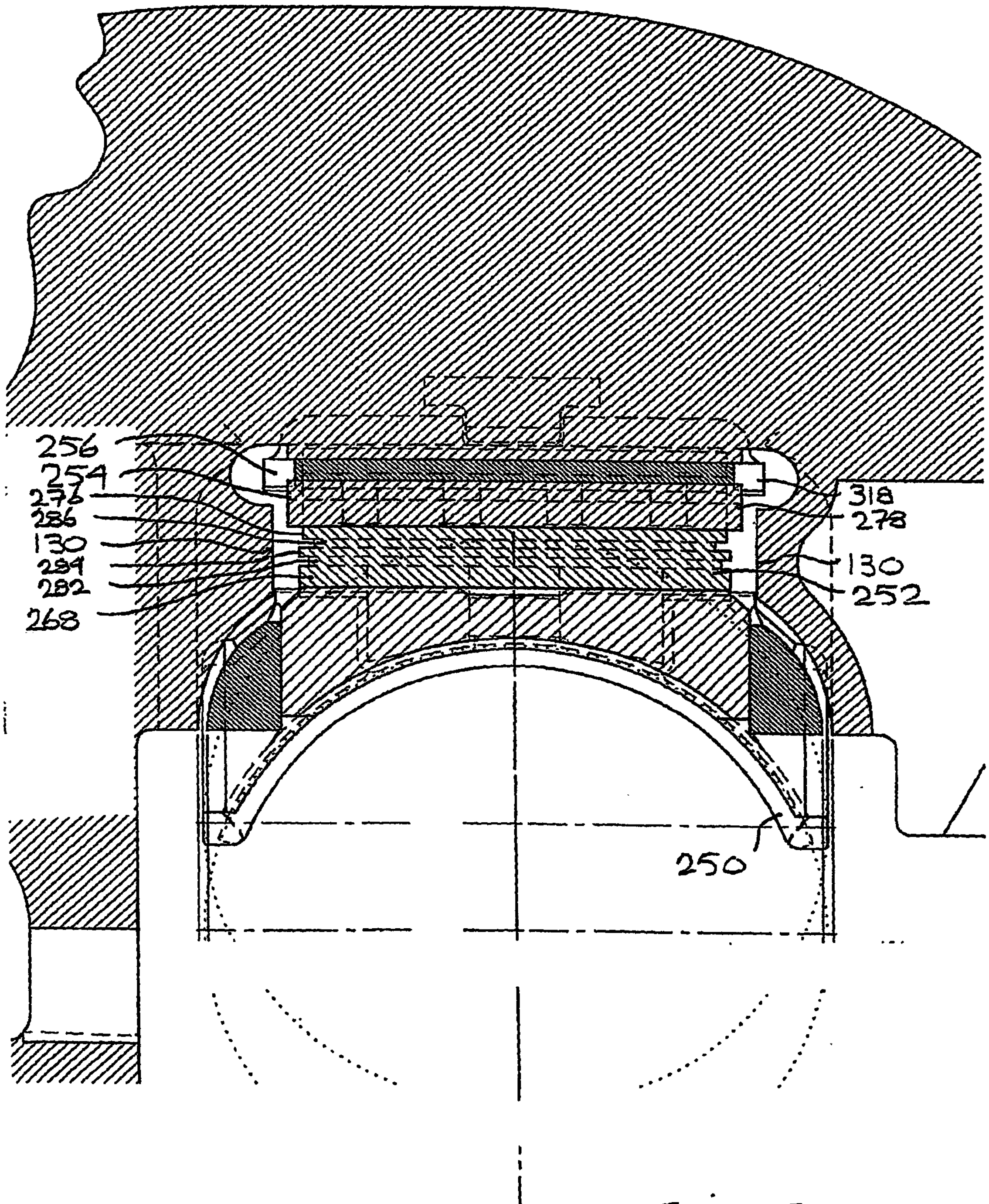


FIGURE 9c.

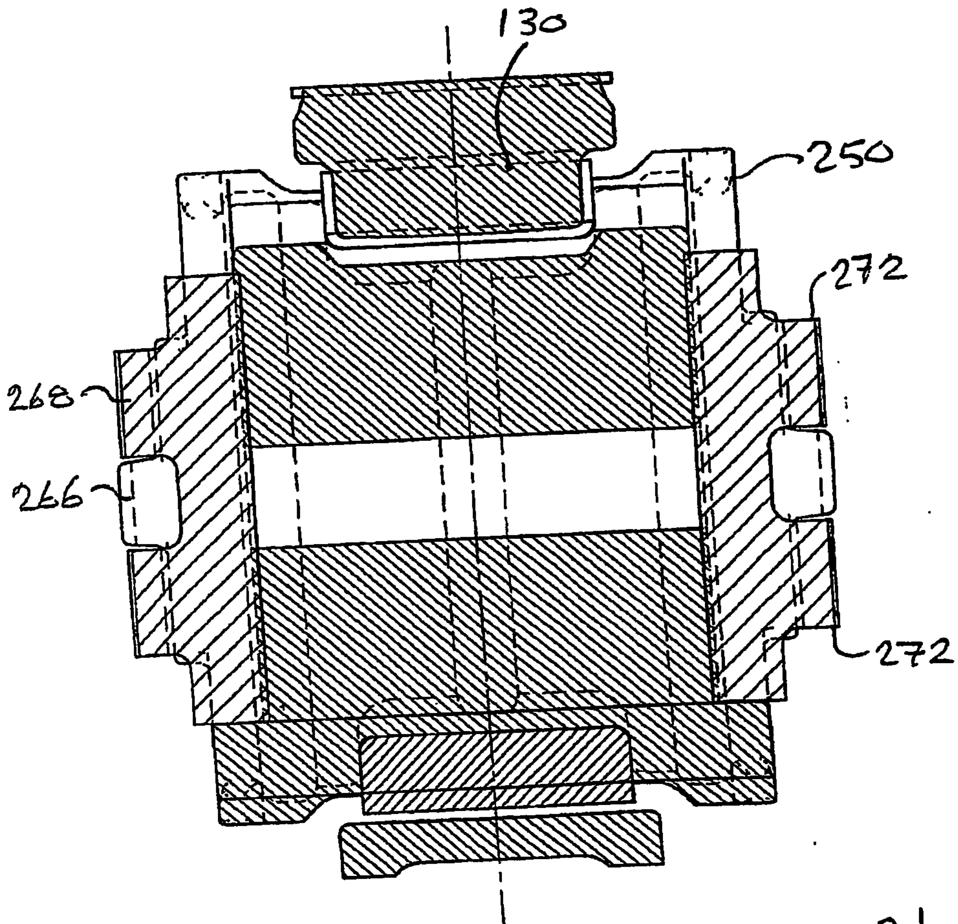


FIGURE 9d

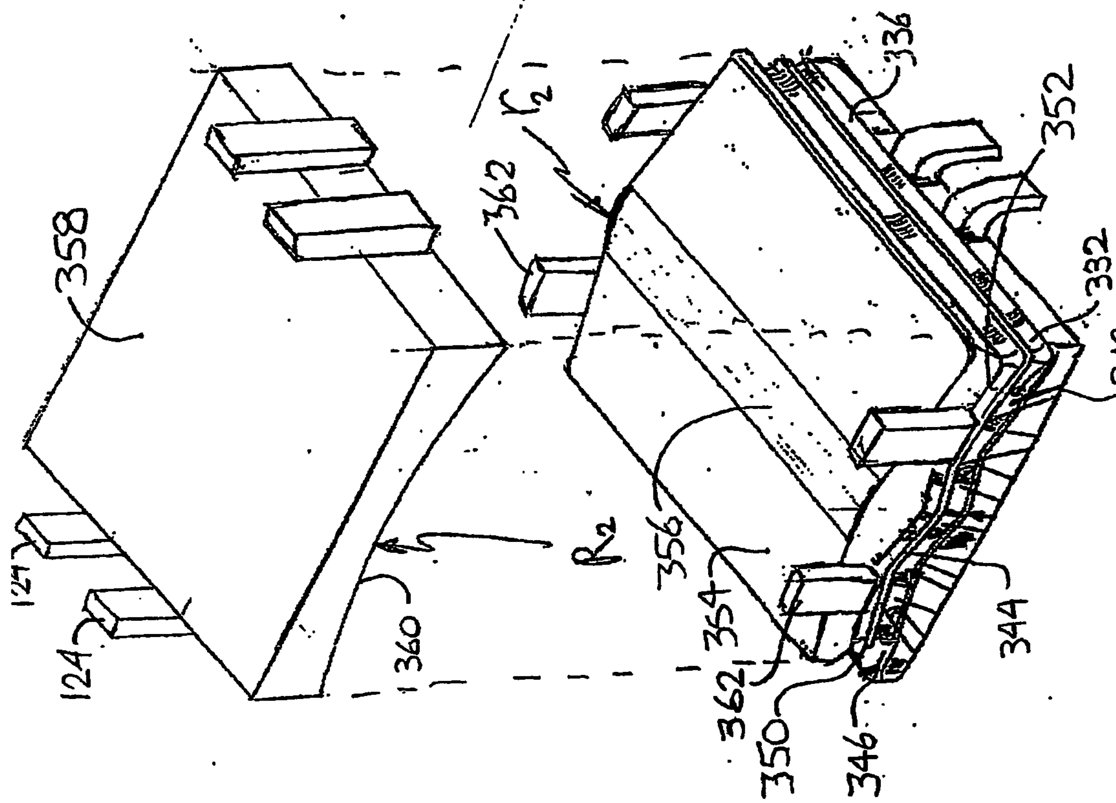


FIGURE 9f

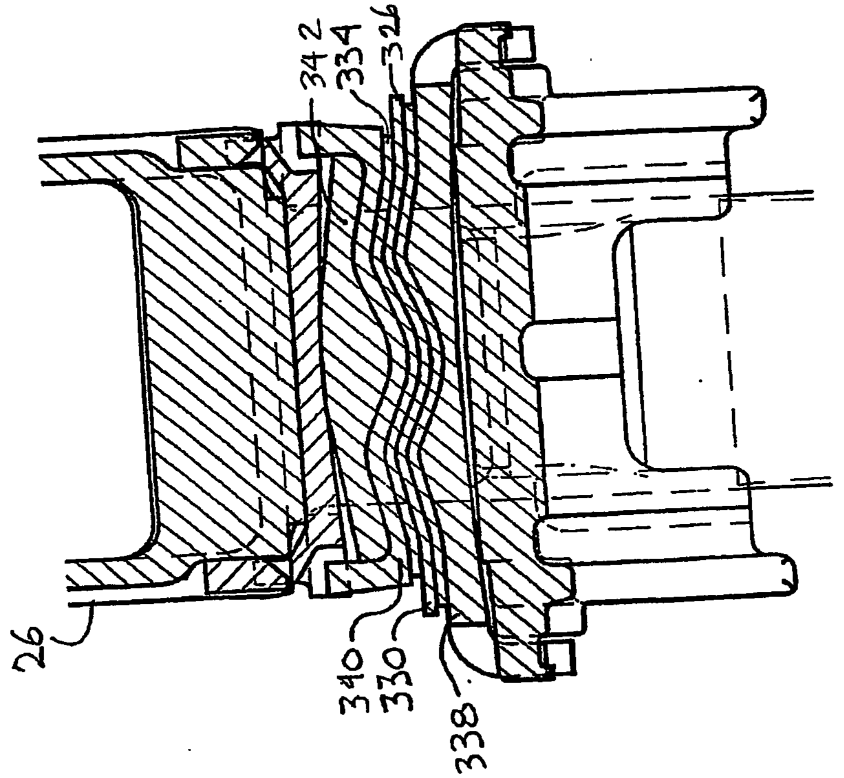


FIGURE 9e

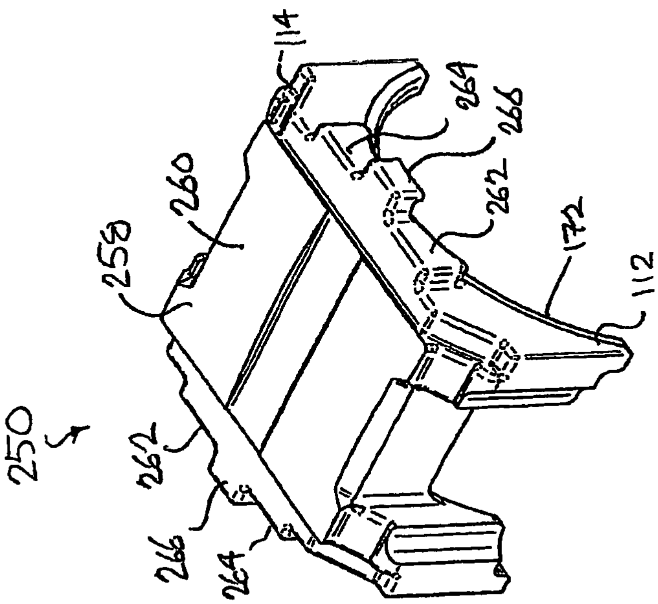


FIGURE 10a

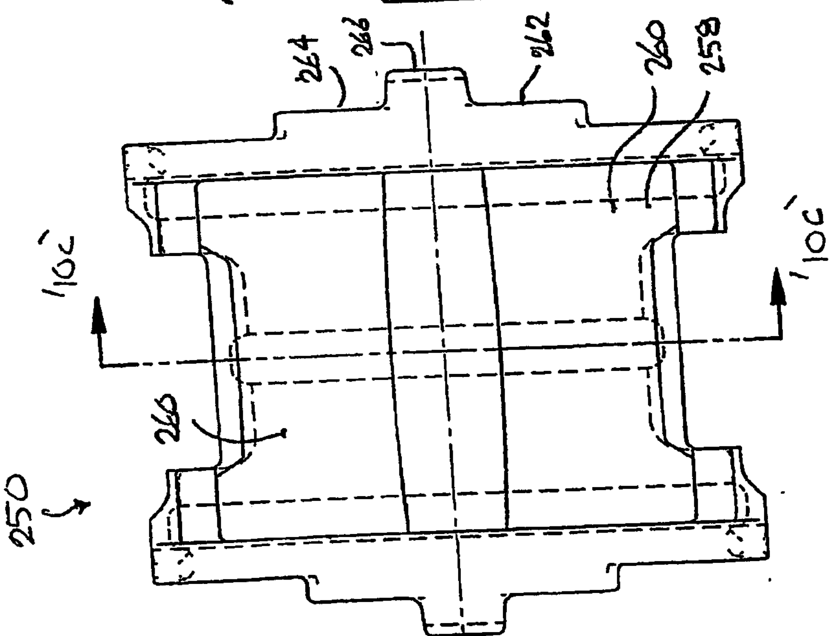


FIGURE 10b.

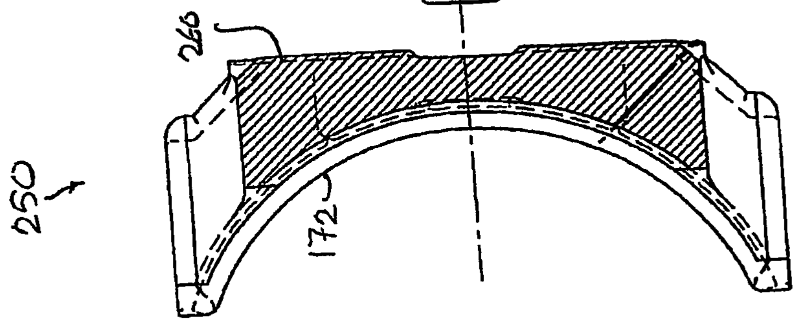


FIGURE 10c.

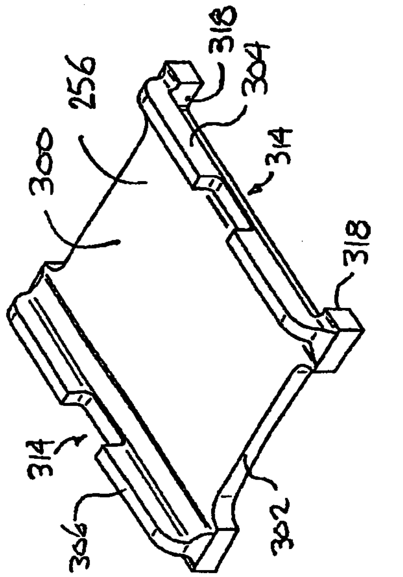


FIGURE 11a

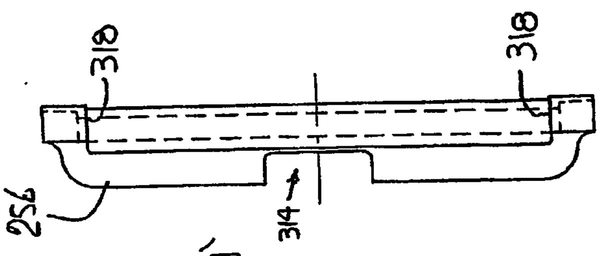


FIGURE 11c

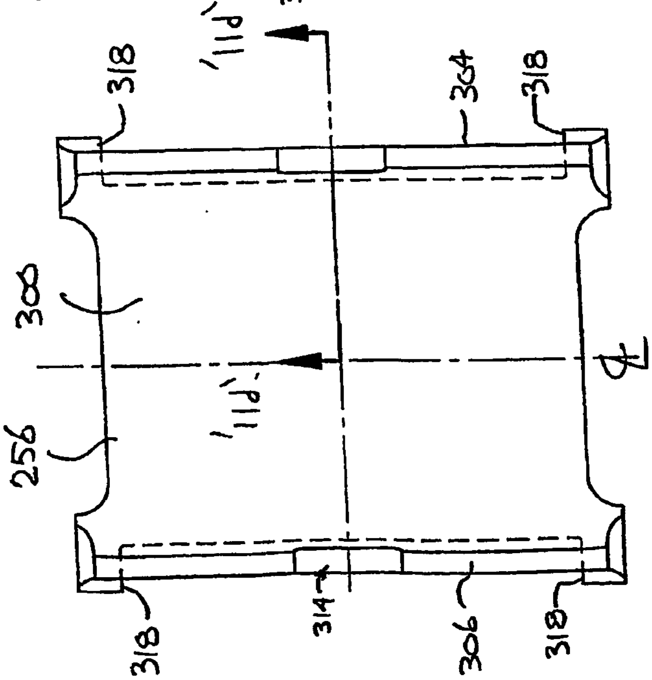


FIGURE 11b

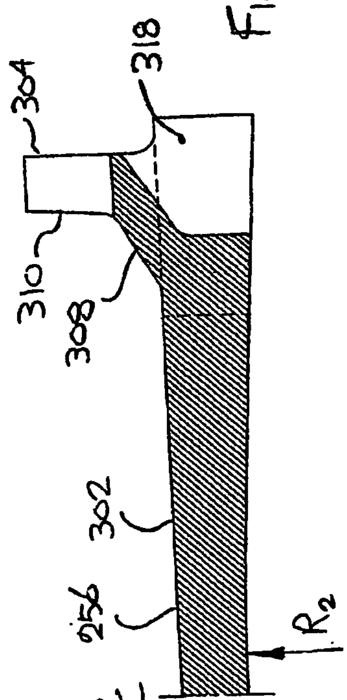


FIGURE 11d.

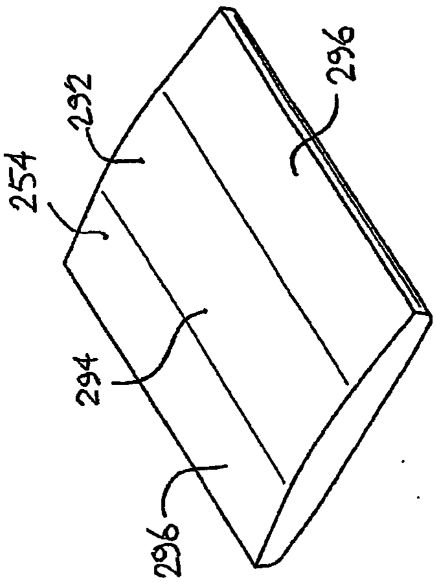


FIGURE 11e

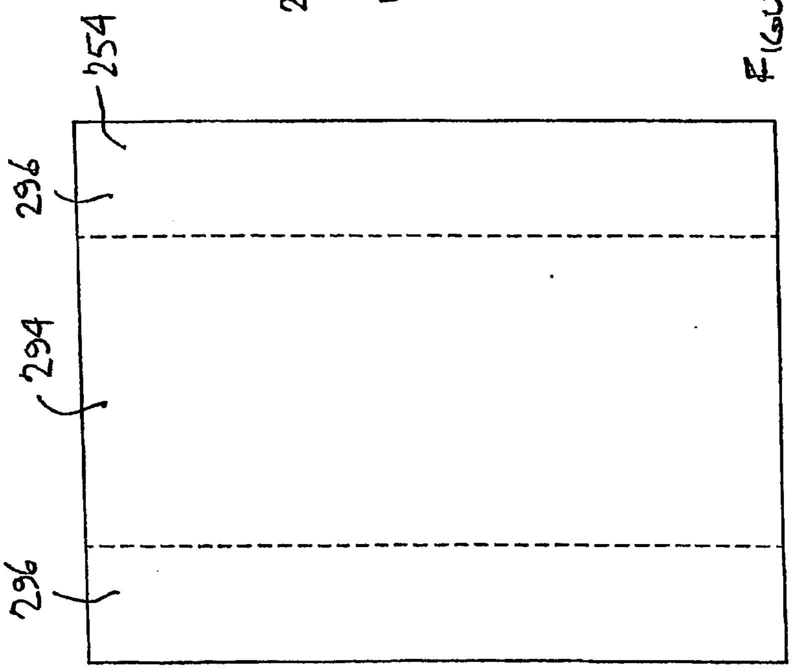


FIGURE 11f

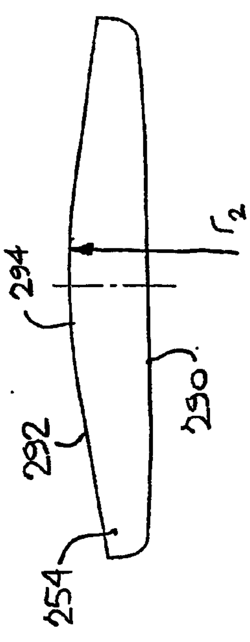
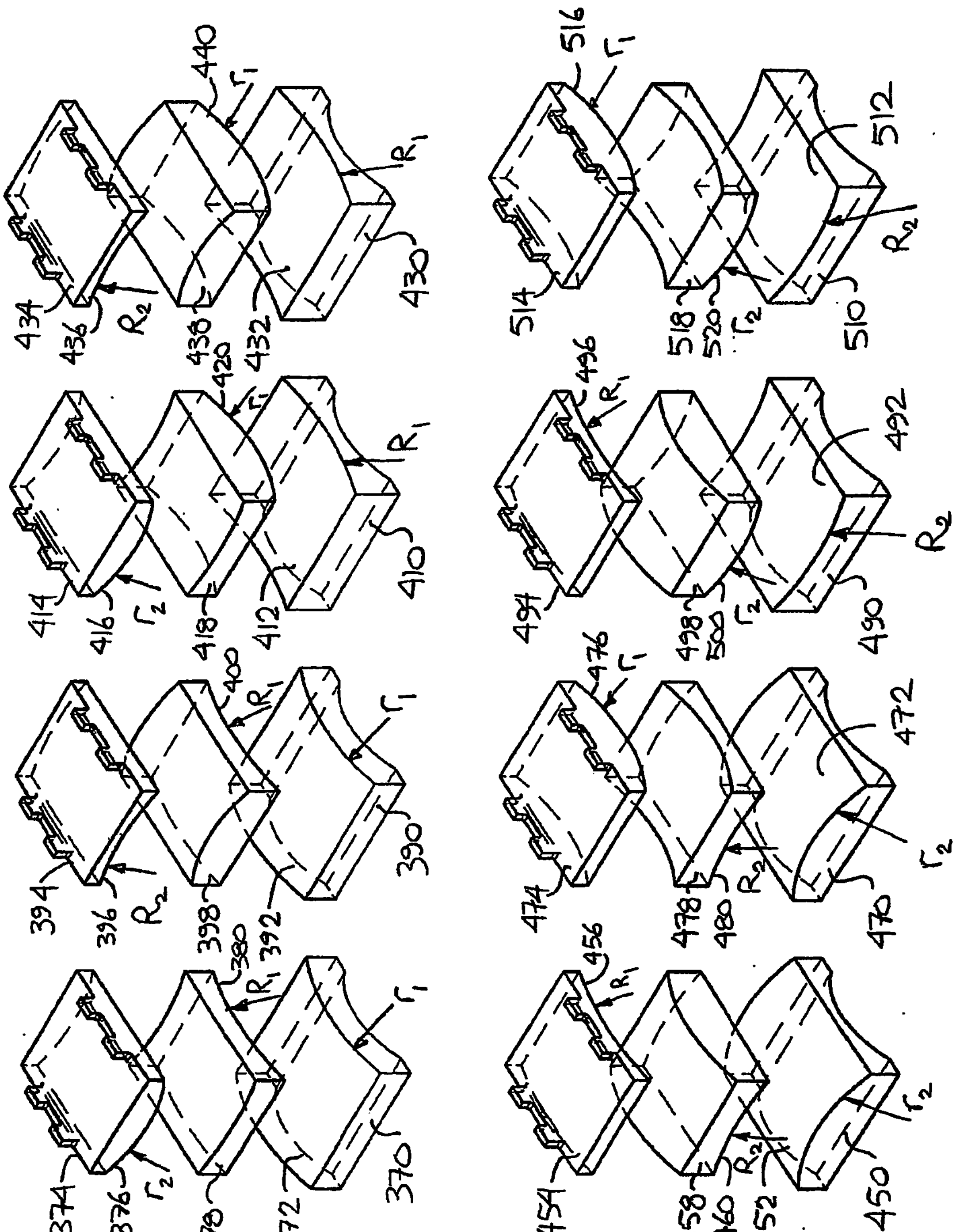


FIGURE 11g.


374
376
r_2
378
372
370
380
R_1
r_1
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396
R_2
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r_1
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416
r_2
418
412
410
420
r_1
R_1
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R_2
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430
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r_1
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r_1
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520
r_2
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512
R_2

FIGURE 12.

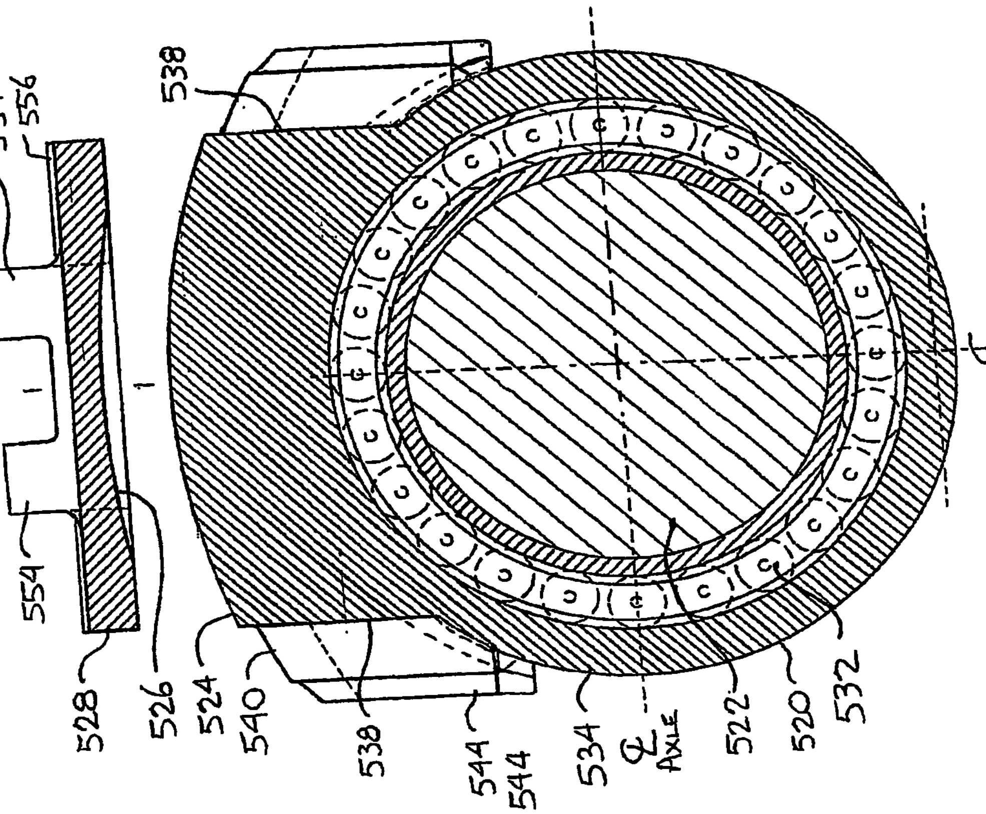

FIGURE 13c

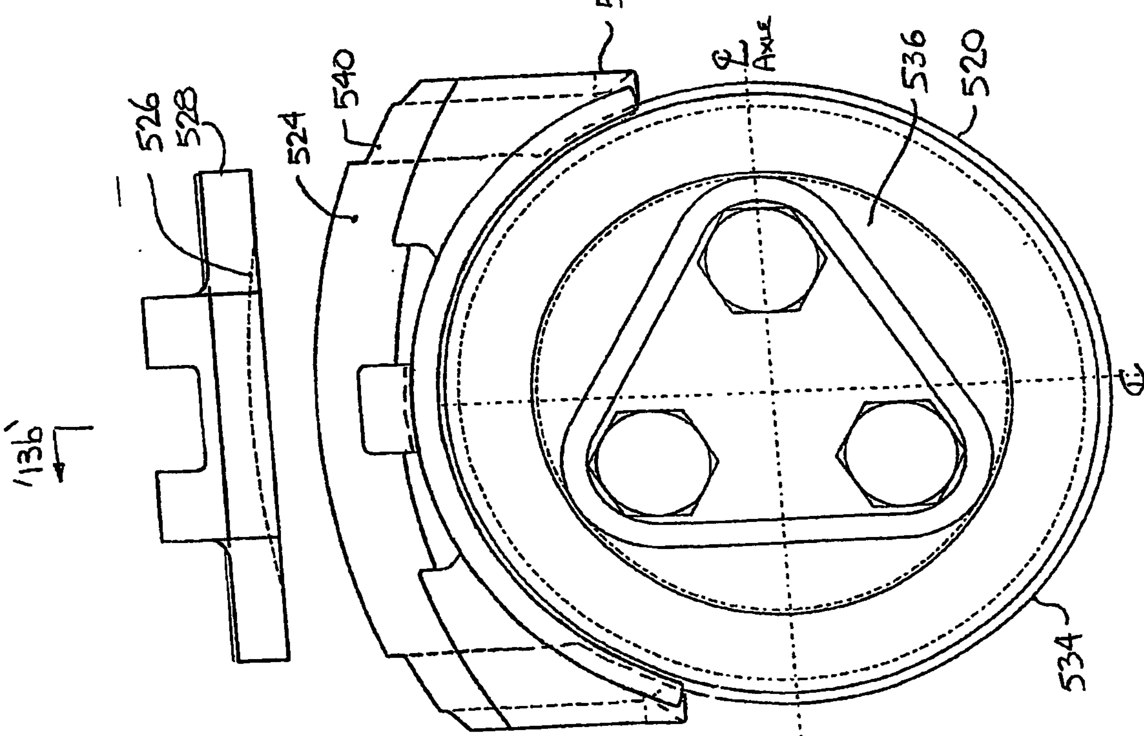

FIGURE 13a

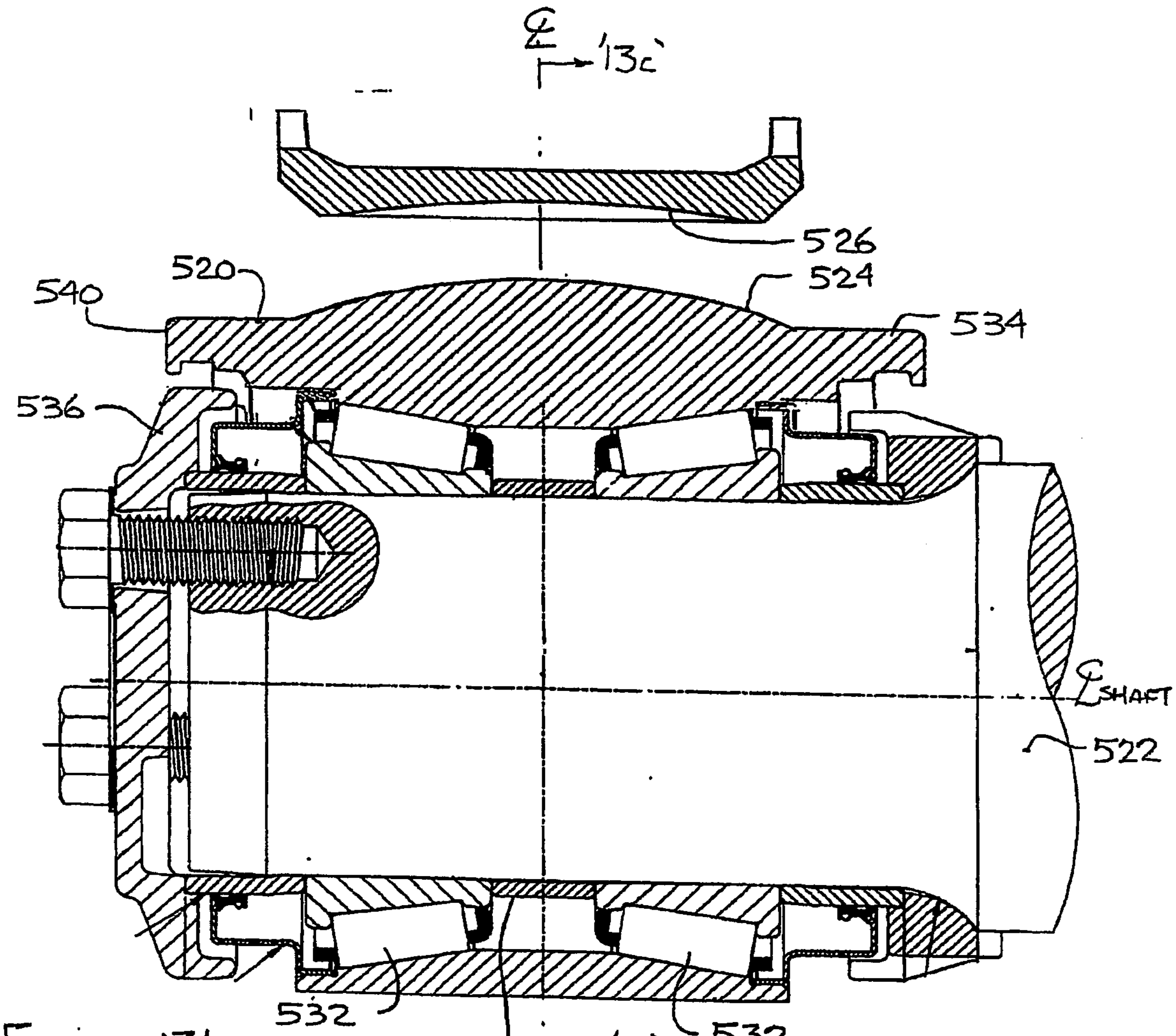

FIGURE 13b

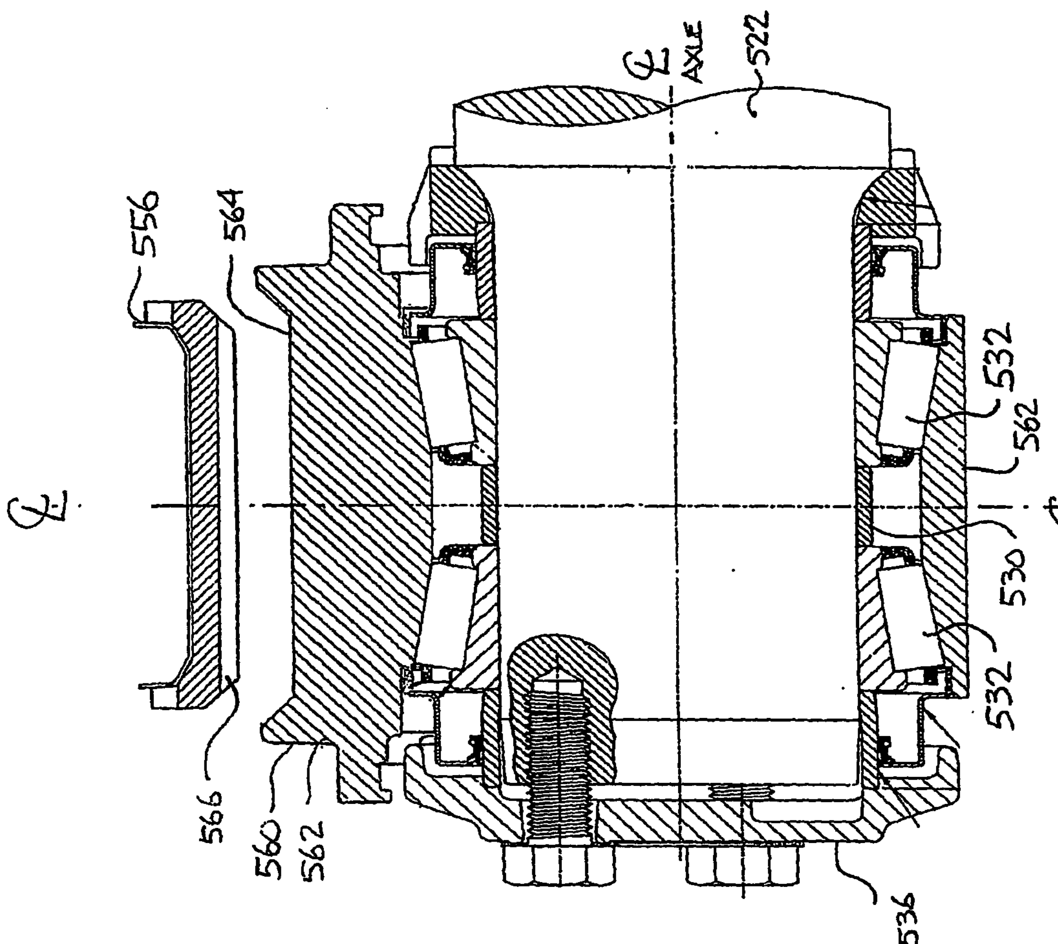

FIGURE 14b

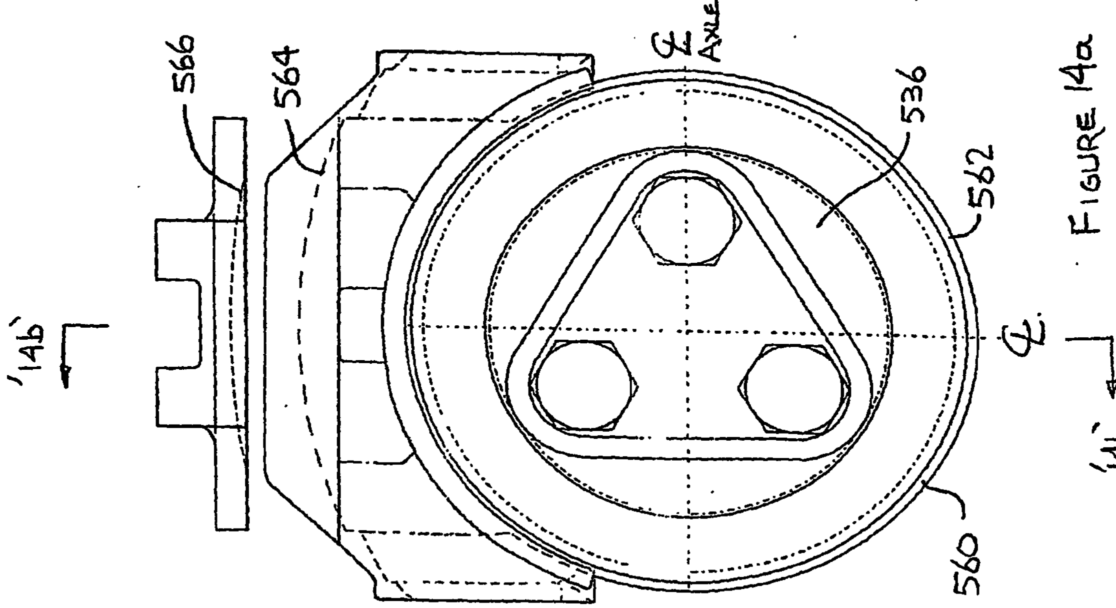

FIGURE 14a